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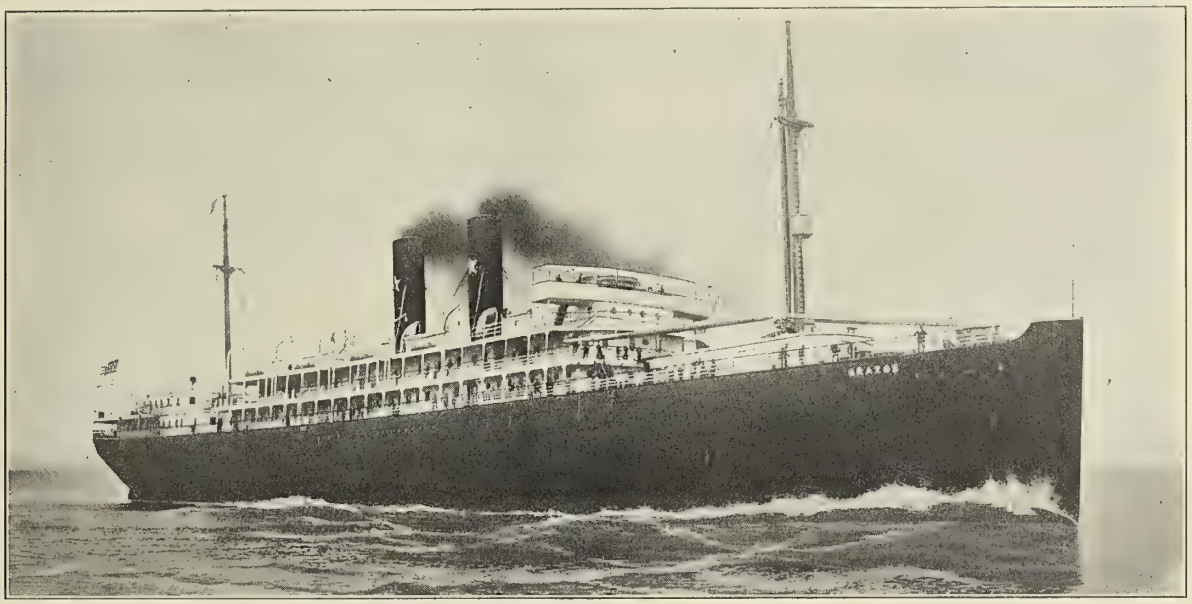
THE MALLORY LINE STEAMSHIP BRAZOS.

BY SIDNEY GRAVES KOON.

This new steel twin-screw steamer has been built by the Newport News Shipbuilding & Dry Dock Company, under the general supervision of Theodore E. Ferris, of Cary Smith & Ferris, New York. The plans were worked up in collaboration with Charles Mallory, and, later, H. H. Raymond, general manager of the line. She sailed from New York to Galveston on her maiden voyage Oct. 3. She was built under special survey of the Bureau Veritas. She is of the hurricane deck type, with straight stern and elliptical stern, and is schooner-rigged, with two steel pole masts.



THE NEW MALLORY LINE STEAMSHIP BRAZOS AT SEA.

The ship has a length over all of 418 feet, a length between perpendiculars of 400 feet, a molded beam of 54 feet, a depth to hurricane deck of 37 feet, to main deck 28 feet 10 inches, and to lower deck 19 feet 6 inches. Her load draft is 24 feet, at which the displacement is about 10,000 tons. The net registered tonnage is 4,077, and gross tonnage 6,399. She is fitted with bilge keels for about half length, consisting of 15-inch plates 5/8 inch thick.

The ship is constructed with a partial double bottom under the boiler space, with ballast and fresh-water tanks for boiler feed of 300 tons capacity; and, in addition, tanks for culinary use of 64 tons and peak tanks of 240 tons capacity for water ballast or fresh water. The arrangement of weights is such that at full load the vessel trims 2 feet by the stern. The total cubic feet of cargo space is about 335,500, equivalent to 10,500 bales of cotton. The bunker capacity is for 1,430 tons of coal. At full-load draft and carrying 1,150 tons of coal, 200 tons of feed-water and 80 tons of stores, passengers and crew, the schedule of weights allows for 4,200 gross tons of freight. With 3,200 tons of freight, and the same provision for coal, etc., the draft is 22 feet. At this draft the sustained sea speed between New York and Galveston was estimated at 15½ knots.

As mentioned, there is only a single bottom outside the boiler room. Five transverse watertight bulkheads to main deck divide the hull into six compartments, but it is noteworthy that the bulkhead between engine and boiler rooms is not watertight. The frames (angles and reverse bars) are spaced 26 inches between centers. Deep frames are fitted on all watertight bulkheads. Web frames, built up of plates and angles, are fitted below the main deck where needed in the machinery space, and are fitted about seven frame spaces apart in the holds. They are also fitted in way of cargo ports between the lower and main and the main and hurricane decks.

The ship has a total of six decks, the lowest being an orlop in the No. 1 hold. There are complete lower, main and hurricane decks; a promenade deck over all deck houses; a boat deck over the amidships deck house. The crown of the deck beams, all of which are of channel section, supported by two continuous girders with wide-spaced stanchions built up of channels, is 12 inches in the total width of 54 feet, while

the tumble-home of the sides is 12 inches at the hurricane deck. The lower and main decks are completely plated, and no wood decks are fitted thereon; a 2½-inch calked wood deck is fitted on the orlop deck beams. On the hurricane deck a wide stringer is fitted to take the coaming plate of the deck house, and the balance of the deck is plated over with 5-pound steel, the whole being covered with a 3-inch Oregon pine calked deck.

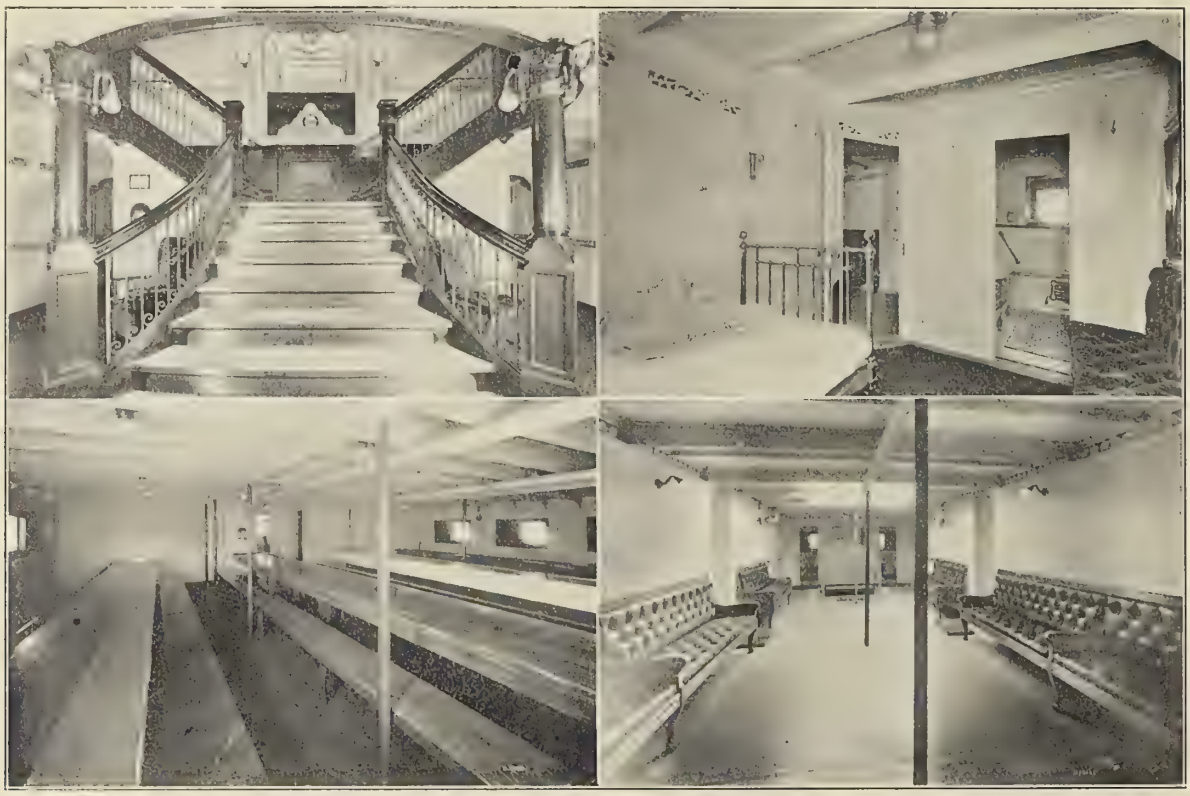
For the storage of cargo there is one hold forward, below the orlop deck; one hold forward of machinery space below lower deck, and one similarly situated aft of machinery space. The 'tween deck compartments include one on the orlop deck; three on the lower deck, of which two are forward, and one aft of the machinery, and compartments forward and aft of machinery on the main deck.

For the passage of cargo to these compartments a number of hatches and cargo ports are fitted. The two principal hatches stretch from side to side of the ship, and are of unusual construction. The side plating between the hurricane and main decks is swung back in the shape of double doors, while the deck plating across the hurricane deck is lifted up in sections, leaving an enormous hole for the reception and passage of cargo. In addition to these two main hatches there are three center-line hatches on the main deck forward of the machinery space and two aft of the machinery space, as well as two small side hatches just aft of the engine room. One of the forward hatches extends up through the hurricane deck. The cargo ports are twelve in number, eight between the main and hurricane decks and four between the lower and main decks. They are evenly distributed forward and aft on each side. For the handling of the cargo two steel pole masts are fitted, each with 8-ton booms, four on the foremast and two on the mainmast. In addition, the foremast has a special boom of 30 tons capacity for the handling of heavy items. Three double-drum winches serve the foremast, while one looks after the needs of the mainmast. The winch for the 30-ton boom is 10 by 12 inches; the three others are each 8 by 8 inches.

The rudder is of the single-plate type, 1 inch thick, with a cast-steel frame; it has a wrought-iron stock 10 inches in diameter, and is hung on five pintles 5 inches in diameter, with brass sleeves. The stern frame is a steel casting. Each shaft is supported by two cast-steel struts.

PROPELLING MACHINERY.

There are two three-bladed propellers, built up, with a cast-iron hub and manganese bronze blades. Each has a diameter of 15 feet with a pitch of 19 feet, the pitch ratio being 1.267.

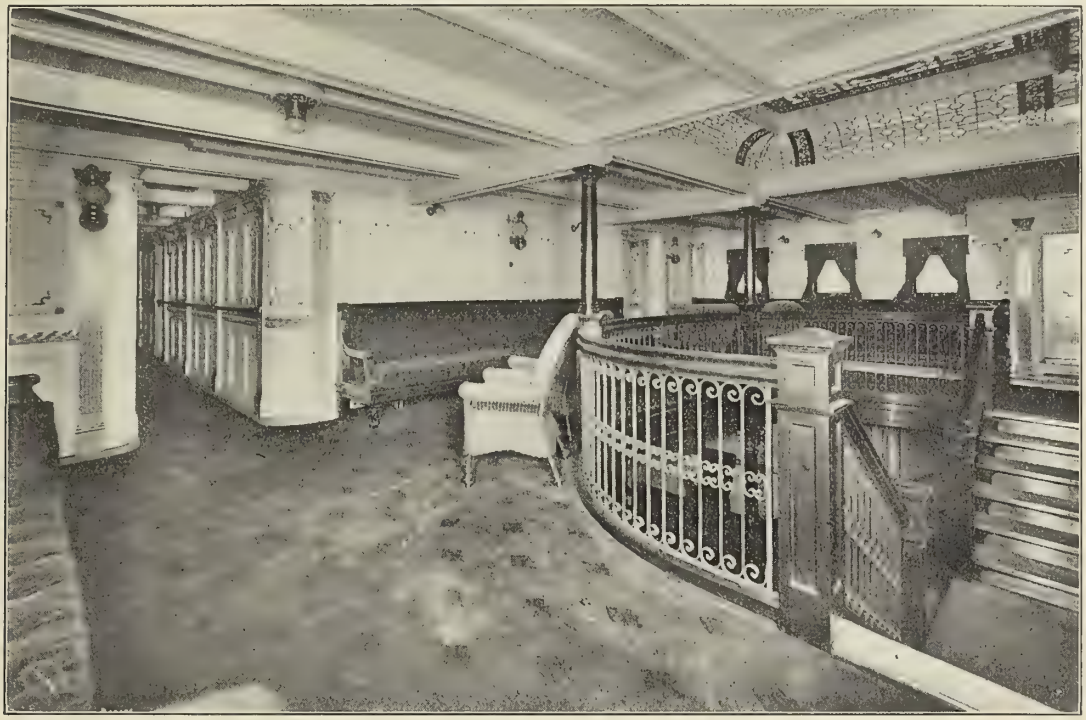


MAIN STAIRCASE.
STEERAGE DINING HALL.

STATEROOM SUITE.
STEERAGE LOUNGING ROOM.

Each propeller is operated by a four-cylinder quadruple expansion engine (both in a broad engine room without divisional bulkheads), with cylinders 23, 33½, 48½ and 70 inches in diameter and a common stroke of 51 inches. These cylinders are arranged with high-pressure forward and the others in regular rotation. All but the low-pressure are fitted with cast-iron liners. The valves are all forward of their respective cylinders, and, except for the low pressure, are piston valves, one being fitted to the high pressure and two to each intermediate. The low-pressure cylinder has a double-ported slide valve fitted with a Lovekin assistant cylinder. All valves have balanced pistons. Stephenson valve gear is fitted, with cast-iron eccentrics, cast-steel straps and forged steel rods and stems. The links are of the double bar type and forged steel; the link block is wrought steel. The exhaust through the low-pressure ports is at about 9,000 feet per minute.

All cylinders have clearances of ½ inch at the top and ⅝ inch at the bottom. The two smaller cylinders of each engine have cast-iron pistons; the other pistons are cast steel. All pistons are fitted with Ramsbottom rings of hard cast iron. The piston rods and connecting rods are of forged steel, the latter being 115 inches between centers. The cross-heads, of



GENERAL VIEW OF THE SOCIAL HALL OR LOUNGE ON THE BRAZOS.

forged steel, have double cross-head bearings. The slippers are cast iron, lined with Parsons white metal.

The reversing gear is of the all-round type with two cylinders at 90 degrees. The turning gear consists of a worm, shaft and a clutch for connection with the reversing engine. The frame is of cast-iron box section for both front and back. The bed-plate is cast iron of a strong box form. The crank shaft in each engine is of two sections built up. The thrust shaft and bearings are of the horseshoe type, with cast-iron body and twelve cast-iron collars or shoes, lined with white metal. The stern tubes are of cast iron with composition bushings; the inboard ends are fitted with composition stuffing boxes.

The engine room extends the full width of the vessel up to the lower deck. Above this, casings extend to the boat deck, where a large skylight is fitted. There are two tunnels for the shafts, and a recess in the after engine room bulkhead for the thrust bearings. This recess, which extends to the lower deck, is fitted with a flat above the thrust bearings, on which are located the electric lighting plant and engineers' work shop.

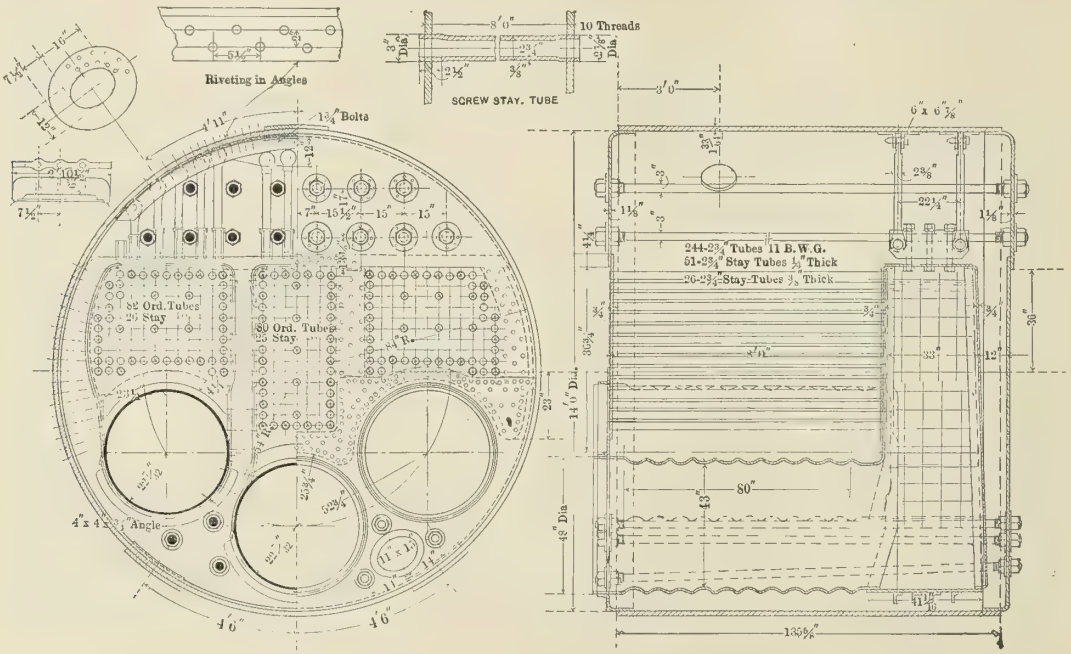
THE SMOKING ROOM ON THE MALLORY LINER BRAZOS.

FIRST CLASS DINING-ROOM ON THE BRAZOS.

BOILERS.

There are eight single-ended return tube Scotch boilers in two stokeholds, located with their grates forward and aft, and one donkey boiler between the uptakes. The main boilers are 14 feet in diameter and 11 feet 9 inches long over heads. Each has three Morison suspension furnaces fitted with Silley patent smoke-box doors. The grate bars are 5 feet 6 inches long. Each furnace has an independent combustion chamber. The total water heating surface is 18,000 square feet, and 4,000 square feet of superheating surface is fitted at the base of the uptake. The grate area is 472 square feet. The ratio of water heating surface to grate is 38.2 to 1; of total heating surface to grate, 46.6 to 1. The working steam pressure is 215 pounds. The exhaust gases and products of combustion pass from the boilers through the superheaters, and then through a nest of air tubes installed for providing hot draft on Howden's system. The two (Foster) superheaters were provided by the Power Specialty Company, New York.

The boiler shells are 1 33/64 inches thick; the back end of combustion chamber, 9/16 inch; back tube sheet, ¾ inch; both heads, ¾ inch. The tubes are 8 feet long, with an outside diameter of 2¾ inches. Each furnace has an inside diameter of 43 inches and a thickness of 21/32 inch. The longitudinal laps of the shell are fitted with double butt straps, treble riveted on each side of the butt. The donkey boiler is vertical, 6 feet diameter and 11 feet 7 inches high. The boilers are lagged with 3 inches of magnesia block, containing 85 percent of pure magnesia. Two funnels are fitted, one to each battery of boilers, and measuring 90 feet high from the base line. Wing bunkers stretch the entire length of boiler space on



STEAM IS FURNISHED BY EIGHT SINGLE-ENDED SCOTCH BOILERS.

each side of the ship, while a cross bunker is fitted forward of the forward boilers. These bunkers are fitted with ventilators, 18 inches in diameter, while the boiler and engine rooms have larger ventilators with large cowls.

AUXILIARIES.

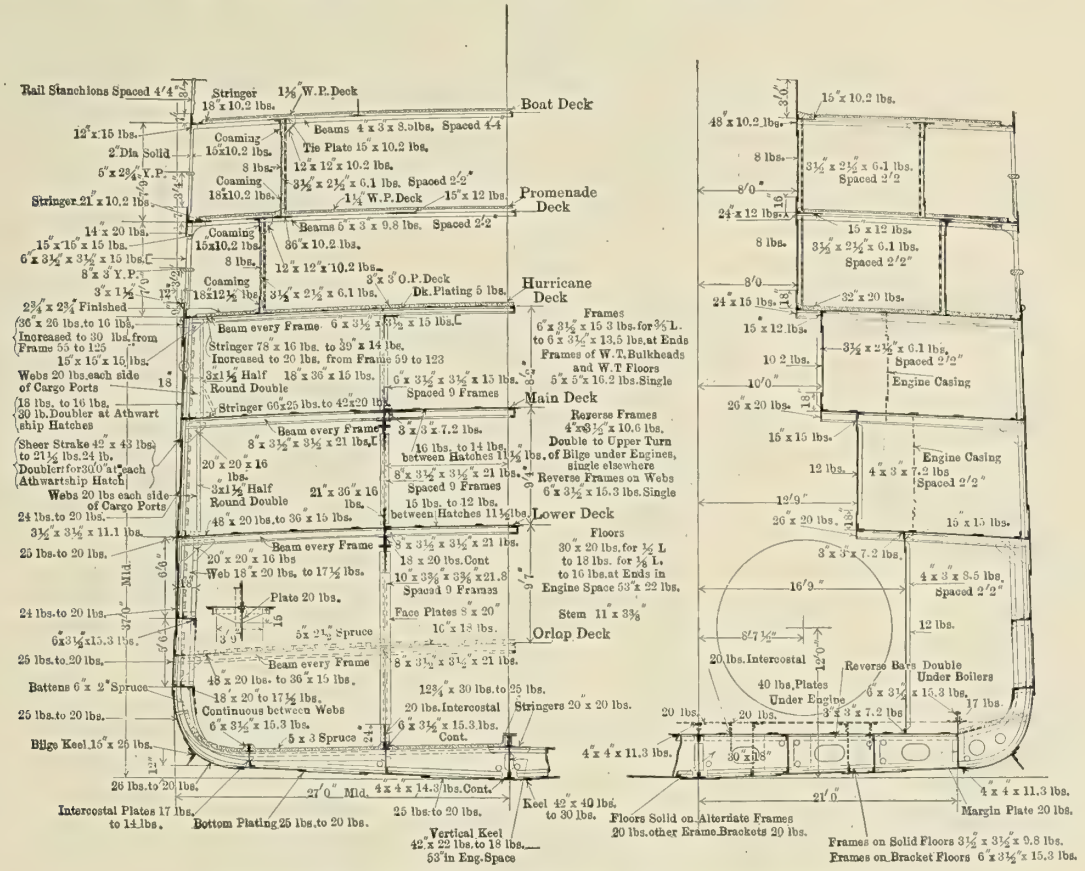
The two main condensers are independent, and are fitted with steel plate shells. They are located in the wings outboard of the main engine. Each has a cooling area of about 5,000 square feet. The auxiliary condenser, with 600 square feet of cooling area, has a cast-iron shell. In both cases the tubes are ¾ inch in diameter. Each main condenser is served by an air pump of the twin-beam independent single-acting type, with cylinders 10 and 22 inches in diameter and a stroke of 18 inches. The two circulating pumps are single-acting centrifugal, with independent engines.

Of the other pumps, all are simplex, and manufactured by M. T. Davidson Company, New York, with the exception of one reserve main feed pump of the Blake duplex type. The main feed is a vertical independent pump, 16 and 10 by 18 inches. Two other pumps of this same size are fitted, one for fire and ash ejector purposes and the other for fire and auxiliary feed. The sanitary pump measures 8 and 8 by 8 inches, and is vertical. Two independent fire pumps are fitted, 8 and 6 by 8 inches. A bilge and fresh-water pump, 4 and 4 by 6 inches, vertical; a donkey boiler feed pump, 12 and 6 by 8 inches, and a combined air and circulating pump for the auxiliary condenser, 8 and 12 by 12 inches, horizontal simplex, complete the list. There is one Reilly feed-water heater, and there are two forced draft fans. The latter are direct connected, with cranks inclosed and running in oil. They were furnished by B. F. Sturtevant Company, Hyde Park, Mass.

Other auxiliaries include two 25-kilowatt Sturtevant electric light engines and generators, each with a cylinder 9 inches in diameter by 8 inches stroke. These run at 350 revolutions per minute, and furnish direct current at a pressure of 115 volts, the maximum load being 217 amperes. The total number of 16-candlepower electric lights installed is about 600; in addition an 18-inch searchlight of 8,000 candlepower is fitted.

The windlass (forward) and two steam capstans (aft) are from the Hyde Windlass Company, Bath, Me. The cargo-hoisting winches were furnished by Williamson Bros. Company, Philadelphia. The steam steering gear was built by the Hyde Windlass Company, and includes a Brown telemotor. The steering gear is fitted right aft, and is of the usual right and left-handed screw gear type. Cast-steel stockless anchors of the Boldt type are fitted. The De Forest system of wireless telegraphy is used, operated from a room on the after promenade deck.

A 60-mile trial trip on the run from the builders' yard to New York was made at a mean draft of about 17 feet. With the engines in full gear and running at about 100 revolutions per minute, the horsepower figured out at about 6,600, and the speed at just over 17 knots. The engines have been so well balanced that during this trial trip the vibration was very slight indeed.



MIDSHIP SECTION AND SECTION THROUGH THE BOILER SPACE SHOWING SCANTLINGS.

PASSENGER ACCOMMODATIONS.

All first class passenger accommodations are located amidships. There are twenty-eight staterooms on the hurricane deck, and seventeen staterooms and six special rooms on the promenade deck. Four of these special rooms are arranged across the forward end of the promenade deckhouse, and are each fitted with brass beds and mahogany dressers. Special combined air ports and ventilators are fitted to these rooms. Two special rooms, with private baths attached, are fitted well forward in the promenade deckhouse, one on each side. Brass beds and mahogany dressers are fitted in these rooms also. In all the staterooms folding metal berths are fitted, and also

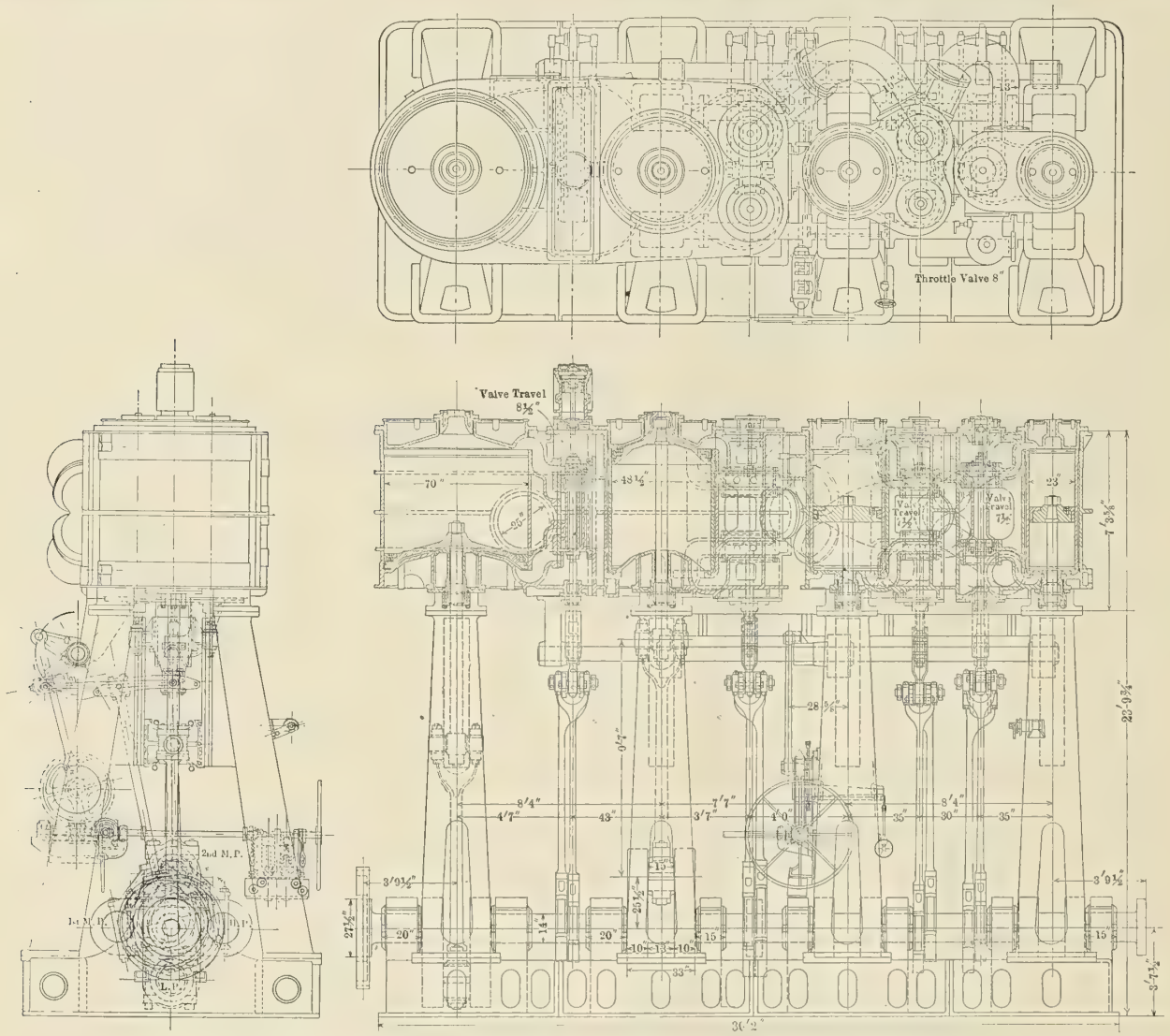
full-length transoms, upholstered in mohair plush, which can be used for berths when occasion demands. Access to all staterooms is from inside passages or from alcoves, there being a continuous fore-and-aft passage on each side in both deckhouses. Divisional bulkheads between staterooms are double in all cases.

The main dining saloon, located in the forward end of the hurricane deck house, is paneled in quartered oak, and is lighted by 20-inch brass air ports, the latter being fitted with inside blinds. In addition a large skylight, with leaded glass dome below, provides light for the dining room and for the interior of the promenade deckhouse. The dining room has a seating capacity for 104 persons.

First class toilet spaces are located between boiler casings on both the hurricane and promenade decks. The decks in these spaces are covered with mosaic tiling, and a tiled wainscoting is fitted all around. Two bath rooms are fitted on each of these decks. The plumbing was furnished by McCambridge & Company, Philadelphia.

The passages and all promenade spaces in the first class passenger quarters on the hurricane deck are paneled in white pine; on the promenade deck composite board with composite ornaments has been used.

Accommodations for the second class passengers are provided in two tiers of steel houses aft. There are thirty-four staterooms, each fitted with three metal berths. There is a



THERE ARE TWO MAIN ENGINES OF THE VERTICAL, FOUR-CYLINDER, QUADRUPLE EXPANSION TYPE.

Several social halls are fitted in the first class accommodations, the principal one being located just forward of the engine casing, where is located the main stairway between the hurricane and promenade decks. This stairway is of mahogany, with metal grills and mahogany top rail. It is lighted through a large skylight with a leaded glass dome below. Writing desks and settees are fitted in the promenade deckhouse in the neighborhood of the light well to the dining room.

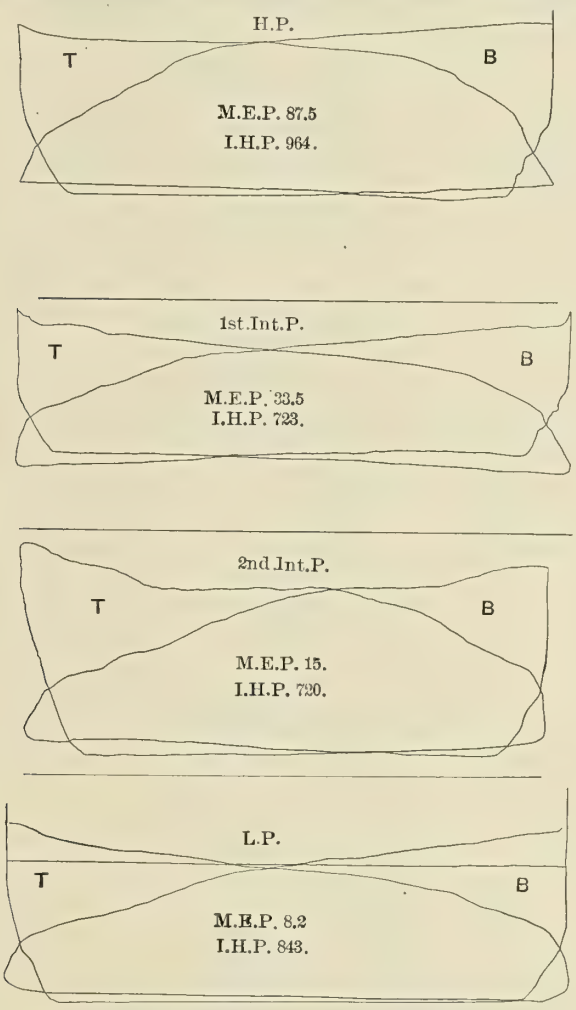
The first class smoking room is located in the after end of the promenade deckhouse, and is paneled in quartered oak. All upholstery in this room is leather of a light tan color. A private inclosed stairway leads from the hurricane deck directly into the smoking room.

large dining room extending the full width of the deck house, and a second class galley aft, fitted with steam kettles, etc.

In the steel house on the boat deck are fitted the captain's stateroom and bath, captain's office, three staterooms for deck officers and an officers' bath room. The captain's quarters are finished in quartered oak. Rooms for the chief engineer and first assistant engineer are located in the promenade deck house opposite the engine casing, to which they have direct access. The chief engineer's room has a private bath attached. Quarters for the engineers, firemen, cooks, waiters, etc., are on the main deck abreast the machinery casing. These quarters are all inclosed with steel bulkheads, and all are fitted with metal berths. Separate mess rooms for engi-

neers and for deck officers are provided. Engineers' toilets and bath, firemen's toilet and showers, and waiters' toilets and showers are located on the main deck in the boiler inclosure. The quartermasters' and seamen's quarters are located on the main deck forward.

The first class galley is located on the main deck forward of the forward boiler casing, and is fitted with all conveniences in use on modern high-class passenger steamers. There is a special inclosure for the bakery. A large ice house, with butcher shop attached, is provided at the sides of the vessel on each side of the galley, and port doors are fitted in the side of the vessel, convenient for filling the ice house and for placing supplies on board. The first class pantry is located on the hurricane deck aft of the dining room, to which it is direct connected by two sets of double doors. It is directly



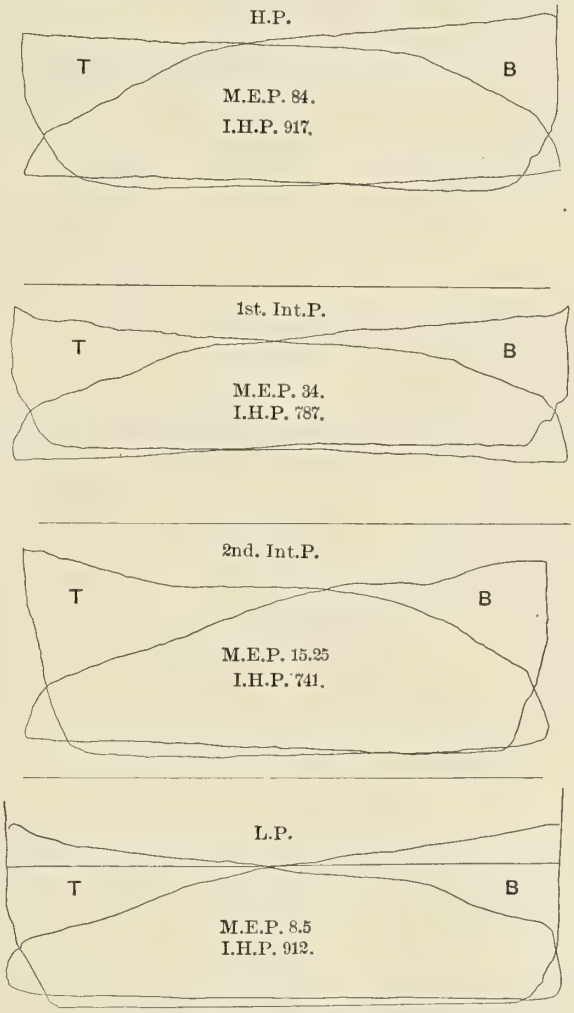
INDICATOR CARDS FROM PORT ENGINE.

over the galley, to which access is obtained by a stairway fitted in the same inclosure; and a dumb waiter is fitted for conveying food from the galley direct. The pantry outfit includes an 11-foot steam table, urns, egg boiler, etc.

During the first round trip between New York and Galveston opportunity was afforded for observing carefully the operation of all the machinery, and for obtaining service trial results. The indicator cards shown were taken October 7, with boiler gage 215 pounds, and a vacuum of 26 inches. The starboard engine, running at 102 revolutions per minute, showed an indicated horsepower of 3,357. The port engine, running at 103 revolutions per minute, showed an indicated horsepower of 3,250. The aggregate was 6,607 indicated horsepower, to which may be added 450 horsepower, assumed for the auxiliaries (see below), making a total of 7,057 horsepower. The indicator cards were taken with springs, measuring, respectively, 100 pounds per inch for the high pressure and 50, 16 and 10 pounds for the other three cylinders, respectively. They were taken under average full-power conditions, and may be said to be fairly representative. The mean effective pressure, referred to the low-pressure cylinder, was 32.67 pounds per square inch starboard and 32.52 pounds port.

The auxiliaries in service, and the estimated horsepower each, are tabulated:

	Horsepower.
Two circulating pumps	194
Two air pumps	41
One feed pump	31
Two blower engines	140
One dynamo engine	41
One sanitary pump	3
Total	450



INDICATOR CARDS FROM STARBOARD ENGINE.

In addition, there were occasionally in service a bilge pump, a fresh-water pump, an ash ejector pump and the steering engine. Some of these, however, were used not more than one hour per day, and their total horsepower may safely be assumed as about 20, making the total estimated horsepower of the auxiliaries about 470.

In computing the coal burned per horsepower per hour, a number of bags of coal were carefully weighed, and in each watch the total number of bags used was carefully recorded. Data obtained in this way, after taking account of the amount of coal put into the ship at New York, and the assumed consumption in port at both places, checked within 5 tons of the bunker measurement on the completion of the round trip. Without going into details in this regard it was estimated that the ship left New York with 1,275 tons in her bunkers.

She burned 520 tons in four and three-quarter days (109 tons per day) when runing at maximum full power to Galveston. Sixty tons were used in port in Galveston, and 450 tons for the return run, which was made under easy conditions.

A number of tests of considerable duration were run, which are given in abstract in the appended table. The mean draft of the ship on leaving port was 20 feet 4½ inches, corresponding with a displacement of about 8,200 tons. The draft on each of these tests was not noted, but must have been less, on account of the continual consumption of coal.

The first test, marked A, was run under full power under natural conditions, with all eight boilers in use, and with the engines in full gear. The second test, marked B, was run under the same conditions as the first. The third test, marked C, was run with six boilers operated at full power, and with the engines in full gear practically all the time. The last test, marked D, was run with six boilers, but the blowers and furnace pressure were cut down somewhat. The engines were linked up, the starboard high pressure 3½ inches, and low pressure 1½ inches; the port, ¼ inch less in each case; the intermediate cylinders were in full gear.

	A.	B.	C.	D.
Duration of trial, hours.....	24	16	40	12
Average speed through the water, knots	16.27	16.39	15.71	15.27
Average revolutions per minute	98.9	99.6	95.44	92.8
Indicated horsepower, main engines .....6291.		6400.	5744.	5372.
Coal per 24 hours, tons.....	90.14	95.9	80.57	79.81
Coal per I. H. P. per hour, main engines, pounds.....	1.337	1.398	1.354	1.37
Coal per I. H. P. per hour, main and auxiliary engines, pounds	1.244	1.303	1.210	1.292

The results, as shown in the consumption of coal per indicated horsepower per hour, are especially fine, the figures given indicating that in no case did the total consumption, based on main engine horsepower alone, exceed 1.4 pounds, while if we include the 470 horsepower of auxiliaries (420 horsepower in the last case) the results are shown to be even better. The slip of the propellers, on a run of 695 nautical miles from Galveston to Dry Tortugas in slack water, and at a mean speed of 16 knots, was found to be 12.2 percent.

MARINE ENGINE DESIGN.

BY EDWARD M. BRAGG, S. B.

Engine Bed.—The stresses to which the engine bed will be subjected are very hard to calculate accurately, as they depend upon the alinement, balance, etc. The thickness of metal necessary for a rigid bed is determined more by experience, and is usually from ⅞ inch to 1½ inches when the bed is of cast iron, and from ¾ inch to 1¼ inches when of cast steel. When made of cast iron, the longitudinal girders and cross girders carrying the main bearings are made of the hollow box form (see Fig. 36), usually open at the bottom, with flanges for the holding-down bolts. When made of cast steel, it is found best to make all parts of the open *I* section, a section consisting of a central web with broad flanges at the top and bottom. (See Figs. 37 and 38.)

The bed is usually made in sections, to facilitate casting and handling, the break in the bed usually coming opposite the coupling flanges of the shaft. The cross girders can be figured as beams supported at the center line of the longitudinal girders, and loaded in the center of their length with the load for which the bearing caps are figured. The width of the cross girders is less than the length of the bearings which they carry, the amount of the overhang of the bearings being about three-fourths of the overhang of the boxes of the crank-pin end of the connecting rod. The sectional area of the cross girder under the bearing should be sufficient to keep the

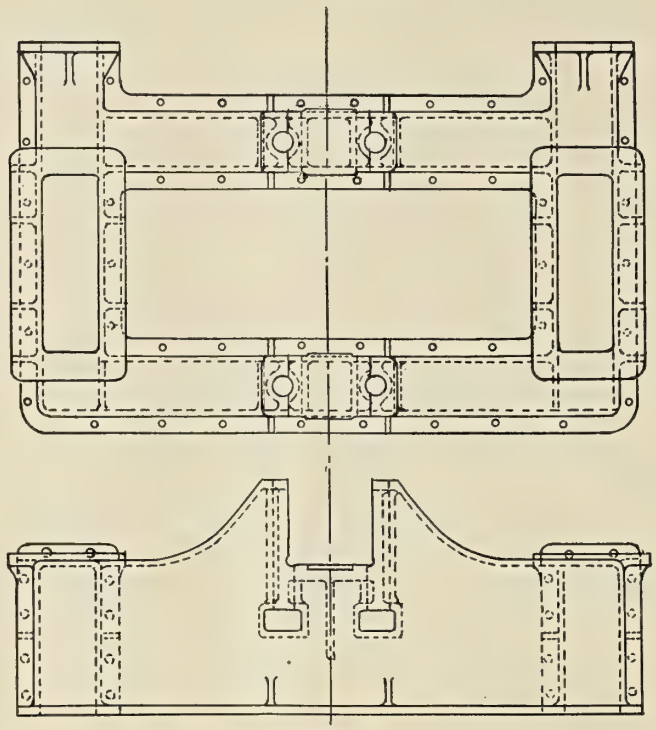


FIG. 36.

stress from 600 pounds to 1,000 pounds per square inch for cast iron, and from 1,000 pounds to 1,500 pounds for cast steel. As the after bearing is subjected to the greatest load, the girder carrying it should be figured, and the depth of the other girders taken from this. This depth will be the minimum depth; that taken finally may be such as will cause the connecting rod to clear the sole plate at the bottom of the stroke.

The distance between the two longitudinal girders should be sufficient to clear the connecting rod as it rotates. To do

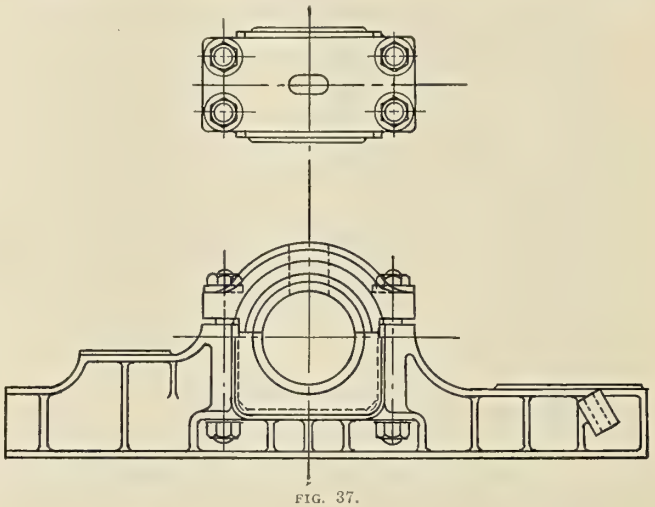


FIG. 37.

this it is best to draw a clearance diagram for the crank-pin end of the connecting rod, showing the path of the extreme points. The side girders should then clear this path by 1 inch to 2 inches. Usually the cross girders are carried from 1½ to 2 inches below the lowest point of the clearance diagram, otherwise the sole plate must be cut away at the crank pits to clear the connecting rod.

Main Bearing Caps.—The main bearing caps are made of cast iron, cast steel, wrought steel or bronze. When made of wrought steel, the brasses are flat backed, as shown in Fig. 39. When made of cast iron, cast steel or bronze, they

are of the shape shown in Fig. 37, and the white metal is put into recesses cored out in the cap. The caps are always provided with hand holes, by means of which the temperature of the shaft can be ascertained; they are also used for applying grease to the shaft. The hand holes are 2½ inches wide and 4½ or 5 inches long, the narrower dimensions being in the direction of the center line of the shaft. The width of the cap will be the same as that of the cross girder carrying the bearing, or from 1 inch to 3½ inches less than the length of the bearing. The load to be used in figuring the bolts and cap should be the maximum upward load coming upon the cap. In the after bearings of the different cylinders the greater loads act upon the top and bottom of the bearings, while in the forward bearings they act upon the sides of the bearings. Caps will be figured, however, as if the maximum loads acted upon the tops of all the bearings. The following factor, applied to the mean loads obtained from formula 44, will give the maximum loads coming upon the bearing caps:

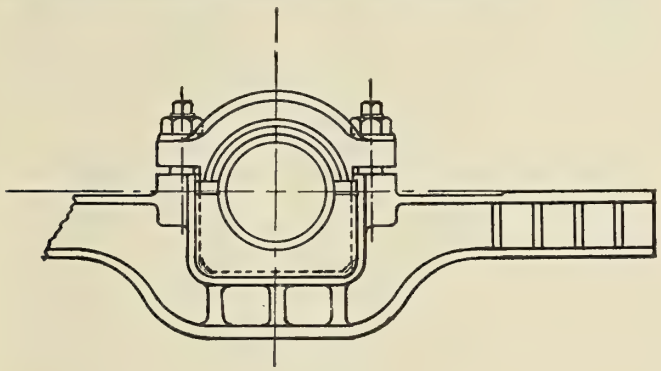


FIG. 38.

Cylinder.	Type of Engine.	Which Bearing.	Factor.
1st	All	Forward and after	2.
2d	Triple	Forward and after	1.75
2d and 3d	Quadruple	Forward and after	1.75
3d	Triple	Forward	1.67
4th	Quadruple	Forward	1.67
3d	Triple	After	1.4
4th	Quadruple	After	1.4

The size of the bolts to carry these loads can be taken from Table IV. The number of bolts in the caps depends upon the construction of the bed. If the bed is of cast iron, two bolts are usually put into each bearing to hold down the cap (see Fig. 36), unless the cap is long, when four bolts will be used. If the bed is made of cast steel, and the cross girders are of the open I section, four bolts are commonly used in order to clear the central web (see Fig. 37). Sometimes a boss is cast in the central web, and two collar studs used, as shown in Fig. 38. When the construction shown in Fig. 36 is used, the studs can be placed close to the shaft, allowing a clearance of ½ to ¾ inch. When the construction is as shown in Figs. 37 and 38, the clearance will be much more.

The caps should be figured as beams supported at the ends and the load distributed over a part of the length. Using the maximum load for which the bolts were designed, the bending moment will be $\frac{Wl}{6}$, where W = load and l = distance between the center lines of the bolts. In obtaining the moment of inertia of the section of the cap, the breadth to be used should be the breadth of the cap minus the breadth of the hand hole, usually 2½ inches.

The height of the cap will be:

$$H = \sqrt{\frac{Wl}{fB}}, \qquad (45)$$

where W = the maximum load upon the cap;
l = the distance between the center lines of bolts;
B = the net breadth of cap;
and f = such a stress as will give a factor of safety of 10, since the caps may be subjected to pounding.

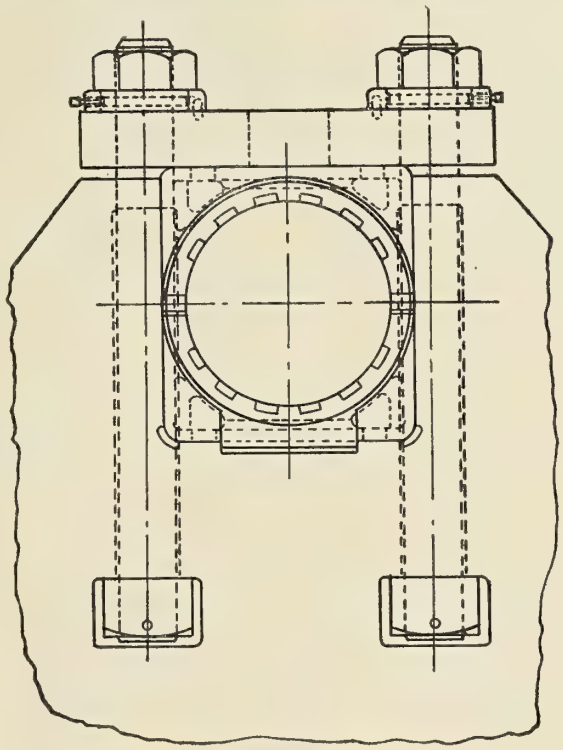


FIG. 39.

If the top of the cap is cored out, the formula

$$\frac{BH^3 - bh^3}{12}$$

should be used in getting the moment of inertia of the section,

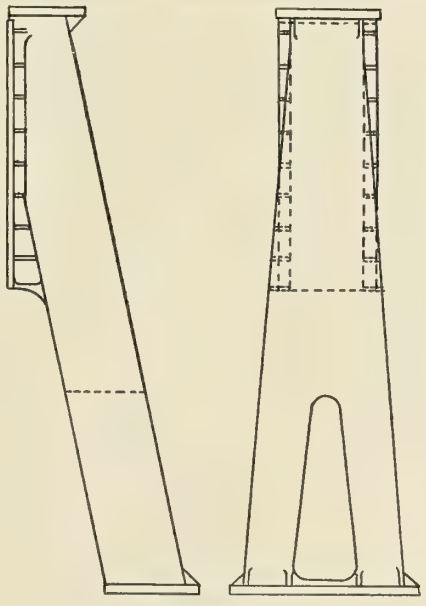


FIG. 40.

where b is the breadth and h is the height of the cored space. In this case the cap will be figured from the formula:

$$f = \frac{WlH}{BH^3 - bh^3}. \qquad (46)$$

ENGINE FRAME.

The engine framing is made of cast iron, cast steel and wrought steel. When made of cast iron, the framing usually consists of hollow inverted Y-shaped supports (see Fig. 40), sometimes called housings, carrying the guide surface for the slipper. When made of cast steel, they sometimes have the same shape as the cast-iron housings, but are made in two parts joined together by a vertical flange (see Fig. 41). The other form in which cast steel is used for engine framing is shown in Fig. 42. These columns are very often used upon the front of the engine, two to each cylinder,

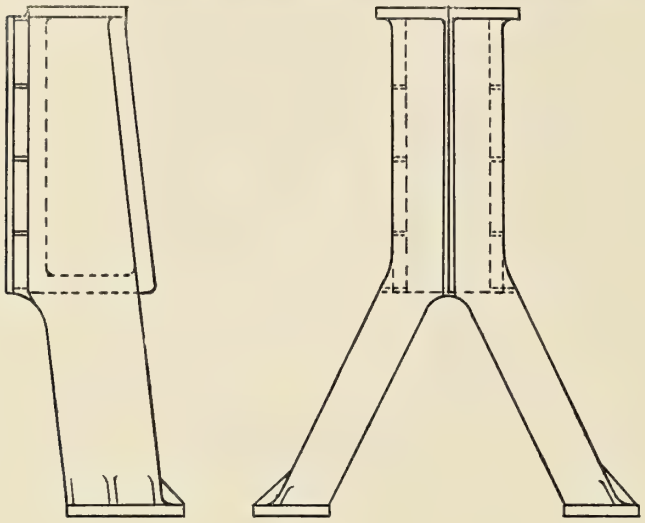


FIG. 41.

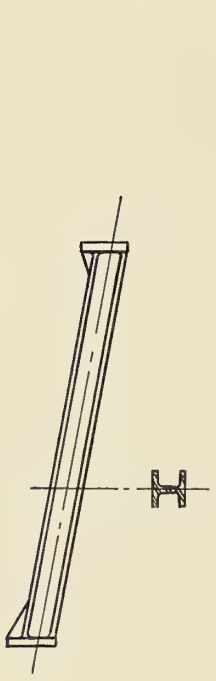


FIG. 42.

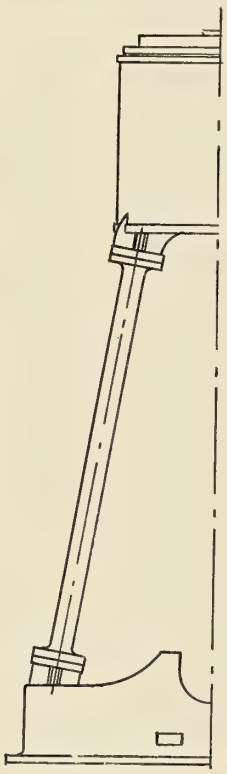


FIG. 43.

in order that access may be had to the crank shaft. When a cross head of the type shown in Fig. 20 is used, four of these columns are used for each cylinder, with a narrow guide surface upon each column. Cylindrical columns of wrought steel are also used, as shown in Fig. 43. When the framing is of wrought steel it consists of cylindrical columns tied and braced longitudinally and athwartship to make the framing rigid (see Fig. 44).

The framing may consist of housings on both front and back of engine, of housings on the back and columns on the front, of cast steel columns on both front and back of engine, or of steel framing. When housings are used, they should be designed to have sufficient area in the smallest section to carry the maximum load coming upon the piston rod, with a

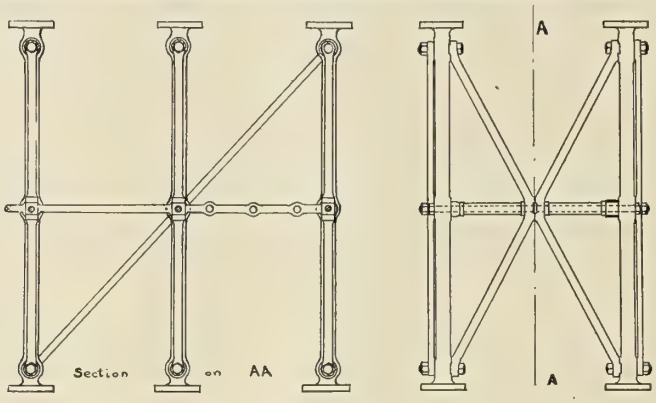


FIG. 44.

stress of 600 to 1,000 pounds for cast iron, and 1,000 to 1,500 pounds for cast steel. The shape of the cross section of the housings will be determined by the size of the slipper guide, and the shape of the flange at the top necessary to accommodate the number of bolts needed to carry one-half of the load on the piston rod.

The columns can be figured by formulæ 21 and 22, given for piston rods. The character of the load is alternating, so that a factor of safety of 12 should be used. Since its length will be considerably greater than that of the piston rod, and its load only one-half of that, its diameter generally comes about the same as that of the piston rod. The column is very often made hollow, to save weight. Whenever it can be arranged, cross-ties are carried from the top of the housing to the foot of the columns, so as to give the housing additional stiffness. When the framing is made entirely of steel columns and ties the stresses are determined by means of a graphical analysis.

Calculations.—The diameter of the crank shaft and diameter and length of the crank pin have already been determined.

$$D = 12 \text{ inches};$$
$$C = 12.5 \text{ inches};$$
$$B = 14'' + 0.25'' = 14\tfrac{1}{4} \text{ inches}.$$

The crank shaft will be made built up, in three interchangeable sections.

$$A = 0.65 \times 12 = 7.8 \text{ inches (use 8 inches)};$$
$$E = 0.28 \times 12 = 3.36 \text{ inches (use } 3\tfrac{1}{4} \text{ inches)};$$
$$F = 0.25 \text{ inches};$$
$$J = 2 \text{ inches};$$
$$K = 12.75 \text{ inches};$$
$$L = 13 \text{ inches};$$
$$M = 11.375 \text{ inches};$$
$$N = 12 \text{ inches}.$$

Assume radius of pitch circle of coupling bolts $= 12'' \times 0.75 = 9'' = r$.

Assume n to be 6.

Formula (43): $$R = \frac{12}{2}\sqrt{\frac{12}{6 \times 9}} = 2.83 \text{ inches}.$$

Let the taper of the bolt be 1 inch per foot, and the length of bolt $= 6\tfrac{1}{2}$ inches, the large end projecting $\tfrac{1}{4}$ inch beyond the flange.

Diameter of bolt at large end

$$= 2.83 + \frac{3.5}{12} = 3.122, \text{ (use } 3\tfrac{1}{8} \text{ inches).}$$

The smaller end of the taper will be

$$3.125'' - \frac{6.5}{12} = 2.583 \text{ inches diameter.}$$

The threaded part on the end can be made 2¼ inches diameter.

A 2¼-inch nut is about 4 inches across the angles, so the pitch circle of the coupling bolts must be at least 2 inches from the outside of the shaft; 9 inches — (6 inches + 2 inches) = 1 inch clearance between nut and shaft. This can be made less if necessary. Try $r = 8.625$ (formula 42).

$$R = \frac{12}{2}\sqrt{\frac{12}{6 \times 8.625}} = 2.89 \text{ inches;}$$

$$2.89 + \frac{3.5}{12} = 3.182'', \text{ (use } 3\tfrac{1}{4} \text{ inches).}$$

Diameter at smaller end of taper $= 3.25 - \frac{6.5}{12} = 2.708$ inches.

The diameter of threaded part can be 2½ inches. The nut will be about 4½ inches across angles; 8.625 — (6 + 2.25) = 0.375 inch clearance between nut and shaft. Diameter of flange = 2(8.625 + 3) = 23.25 inches.

Loads Upon Bearings.—Values of f from Fig. 31 for 3,000 *I. H. P.*, built-up shaft, give the following loads upon the main bearings:

First cylinder, forward

$$L = \frac{21{,}000}{850} \times 0.77 \times 1{,}000 = 19{,}000 \text{ pounds.}$$

First cylinder, aft

$$L = \frac{21{,}000}{850} \times 0.77 \times 1{,}000 = 19{,}000 \text{ pounds.}$$

Second cylinder, forward

$$L = \frac{21{,}000}{850}(1{,}000 + 0.11 \times 1{,}000) = 27{,}400 \text{ pounds.}$$

Second cylinder, aft

$$L = \frac{21{,}000}{850}(1{,}000 + 0.61 \times 1{,}000) = 39{,}800 \text{ pounds.}$$

Third cylinder, forward

$$L = \frac{21{,}000}{850}(2{,}000 - 0.14 \times 1{,}000) = 46{,}000 \text{ pounds.}$$

Third cylinder, aft

$$L = \frac{21{,}000}{850}(2{,}000 + 0.64 \times 1{,}000) = 65{,}200 \text{ pounds.}$$

Area for each high-pressure bearing

$$= \frac{19{,}000}{200} = 95 \text{ square inches,} \quad \frac{95}{12} = 7.92''$$

Area for second cylinder, forward

$$= \frac{27{,}400}{250} = 110 \text{ square inches} \quad \frac{110}{12} = 9.16''$$

} use 9 inches.

Area for second cylinder, aft

$$= \frac{39{,}800}{275} = 145 \text{ square inches} \quad \frac{145}{12} = 12.1''$$

Area for third cylinder, forward

$$= \frac{46{,}000}{300} = 153.5 \text{ square inches} \quad \frac{153.5}{12} = 12.8''$$

} use 12½ inches.

Area for third cylinder, aft

$$= \frac{65{,}200}{350} = 187 \text{ square inches} \quad \frac{187}{12} = 15.58''$$

} use 15½ inches.

See Fig. 46 for sketch of the three sections of shafting.

Loads for which main bearing caps and bolts are to be figured are:

High-pressure bearings, $19{,}000 \times 2 = 38{,}000$
Medium-pressure forward bearings, $27{,}400 \times 1.75 = 47{,}900$
} use 48,000 pounds.

Medium-pressure after bearings, $39{,}800 \times 1.75 = 69{,}700$
Low-pressure forward bearing, $46{,}000 \times 1.67 = 76{,}800$
} use 75,000 pounds.

Low-pressure after bearing,

$65{,}200 \times 1.4 = 91{,}250$ pounds.

Main bearing bolts for 48,000-pound load must be two of 2¾ inches.
Main bearing bolts for 75,000-pound load must be two of 3¼ inches.
Main bearing bolts for 91,250-pound load must be two of 3½ inches.

Make lower brass "half-round" with a thickness of $\frac{D}{10}$ + 0.25 inch = 1.2 inches + .25 inch = 1.45 inches (use 1½ inches). Bolts will clear shaft by 2½ inches to allow for the brass and the metal of the bed.

Distance between centers of 2¾-inch bolts
$= 12'' + 5'' + 2.75'' = 19.75$ inches.

Distance between centers of 3¼-inch bolts
$= 12'' + 5'' + 3.25'' = 20.25$ inches.

Distance between centers of 3½-inch bolts
$= 12'' + 5'' + 3.5'' = 20.5$ inches.

Handholes in main bearing caps to be 2½ inches wide.

Caps to be of cast steel, and stress to be $\frac{60{,}000}{10} = 6{,}000$ pounds.

Caps for high-pressure and forward medium-pressure bearings:
Formula (45):

$$H = \sqrt{\frac{48{,}000 \times 19.75}{5.5 \times 6{,}000}} = 5.36 \text{ inches.}$$

Caps for after medium-pressure and forward low-pressure bearings;

$$H = \sqrt{\frac{75{,}000 \times 20.25}{8.5 \times 6{,}000}} = 5.47 \text{ inches.}$$

Caps for after low-pressure bearing:

$$H = \sqrt{\frac{91{,}250 \times 20.5}{10.5 \times 6{,}000}} = 5.45 \text{ inches.}$$

Make all of the caps 5½ inches thick from the back of the white metal, or 6 inches thick, allowing for the white metal.

(To be continued.)

Revision of the United States Laws Relating to the Safety of Life at Sea.

President Roosevelt has recently appointed a commission for the revision of laws relating to the safety of life at sea. One of the most important duties of this commission lies in a careful revision of the laws which relate to the construction and inspection of steam boilers. The rules at present in force are very much out of date, and need thorough revision to meet the requirements covering the best materials and best methods of modern practice in construction; expert inspection in materials, design and workmanship; conformation of the United States rules with the requirements of marine insurance companies; and adequate provision for conditions on the Great Lakes and Mississippi River with its tributaries.

DESCRIPTION OF A 140-TON FLOATING CRANE WITH A TEST CAPACITY FOR LIFTING 200 TONS.

BY H. PRIME KIEFFER.

Practically up to the present time, shear-leg construction has been exclusively employed for floating cranes. With this system the feet of the forelegs lie at the pontoon gunwale and the load must be passed between the legs, an arrangement which has a number of striking disadvantages. In the first

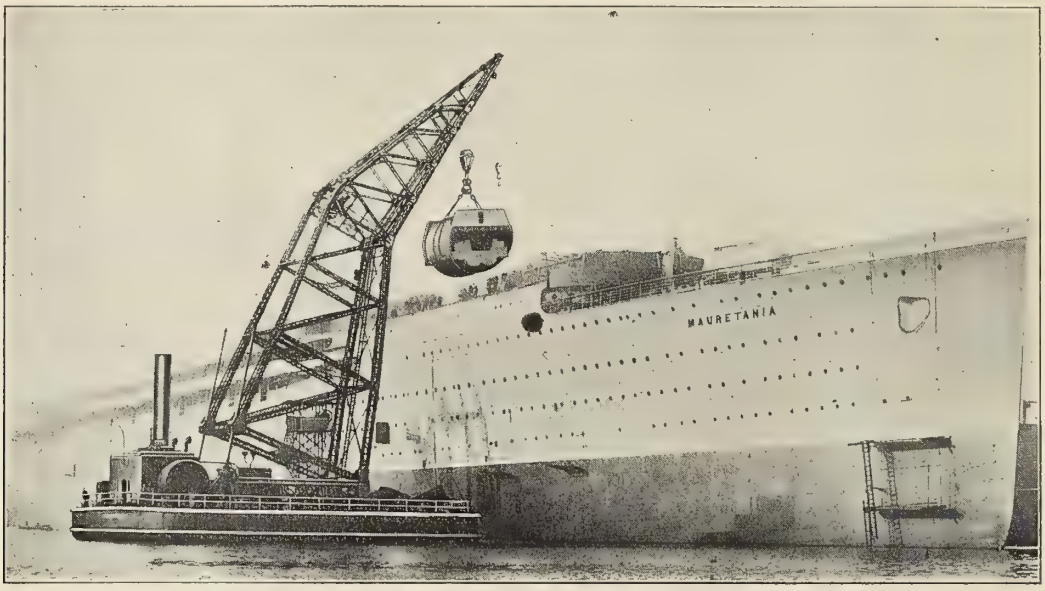


FIG. 1.—140-TON FLOATING CRANE PLACING BOILERS ABOARD THE MAURETANIA.

place the loads to be handled must not exceed certain dimensions, limited, of course, by the sloping position of the fore shear-legs. Secondly, the outreach of the crane from the turning point of the legs to the middle of the hook cannot be used to its full extent, for even at a small inclination the legs will be in touch with the side of high ships. (See Figure 2). Finally, the legs are clumsy and expensive, owing to their length.

These drawbacks are entirely overcome by a new and interesting type of floating crane built by the Duisburger Machinenbau, A. G., formerly Bechem & Keetman, of Duisburg, Germany. The general appearance of this new crane can best

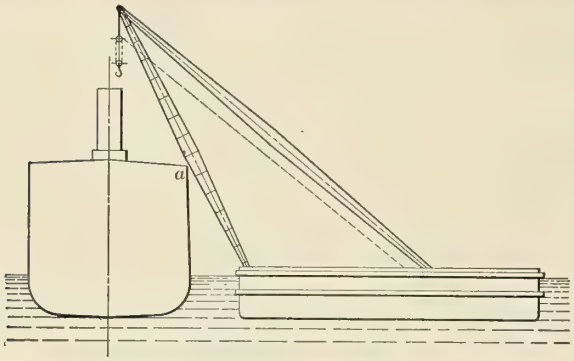


FIG. 2.—ILLUSTRATING THE DRAWBACKS OF A SHEAR-LEG CRANE.

be gained by a glance at the illustrations shown in connection with this article.

In this new construction the turning points of the jib are kept so far back from the pontoon gunwale that sufficient space is left in front of them for taking up the materials to be handled. Thus the latter need no longer be passed between the crane legs, and may, therefore, have any dimensions. Masts, stacks, boilers and other long and cumbersome parts of a vessel may easily be fitted with the help of a crane of this new construction, while this work becomes rather difficult, and very often impossible, when the usual shear-leg crane is employed. By withdrawing the turning point from the foot of the jib we further attain the advantage that the legs do not come into contact with the vessel's side. The crane barge can lie close alongside of the ship, and the jib, as far as the screw spindles permit, may be adjusted towards the outside.

From the constructional point of view this type of crane has the further advantage that the whole jib can be of lattice work, and thus it is of a relatively small weight. In addition to this, the principal parts of the jib can, if desired, take an angular shape and need no longer be led in a straight line.

Several cranes of this type, with a capacity of 100 tons, have

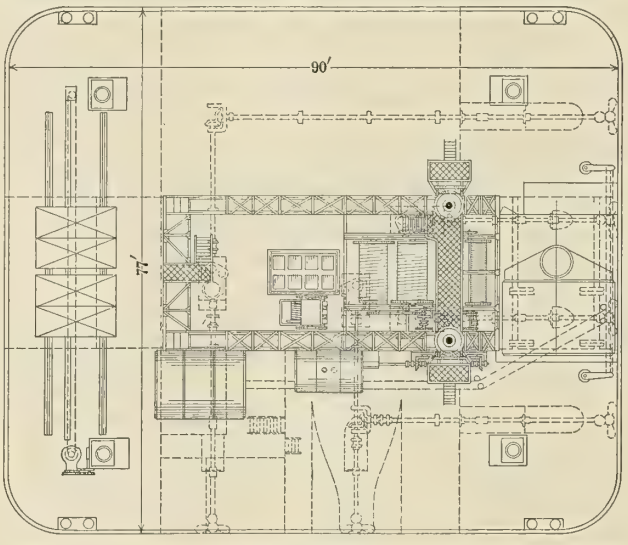


FIG. 3.—PLAN VIEW OF PONTOON, SHOWING LOCATION OF PROPELLING MACHINERY.

been built by the Duisburger Machinenbau, A. G., but not until comparatively recently did they attempt one so large as 140 tons with a test of 200 tons. When Messrs. Swan, Hunter, Wigham and Richardson took the contract to build the mammoth Cunard Liner "*Mauretania*," they realized that the fitting out of the ship would present some serious difficulties, if done by old methods, as some of the equipment, notably the boilers, was very large and heavy. They therefore ordered a crane of the new type with a capacity of 140 tons, which was considered at the time rather high, as such a great capacity had

never before been attempted when building a floating crane.

It must be pointed out that a serious restriction as regards the structure of the crane was imposed upon the constructors on account of the bridge situated near Elswick-on-Tyne. The lower flange of the bridge is in spring tide only about 85 feet above the surface of the water. The crane, however, when erected, has a height of 131 feet, and consequently it was necessary to provide a lowering arrangement, so that the crane could pass underneath this bridge. The jib is therefore swung around two fore foot points and is adjusted by means of two spindles of Siemens-Martin steel. Water ballast serves as a counter weight, the water being placed in two water-tight compartments of the pontoon. The large boiler for the engine also serves as a counterweight.

The square pontoon has a size of 90 feet by 77 feet and a height of 13 feet 10 inches. The corners are rounded. When 140 tons are on board and the load is suspended at the hook, the draft is 8 feet, the inclination 6 degrees and the height from the water surface to the upper edge of the pontoon about 10 inches. Two longitudinal and two cross partitions divide the interior of the pontoon into nine compartments. The four engines are placed in the center room underneath the crane structure. Loads can be transported on the fore deck by means of several truck cars running on rails. Four steam capstans are arranged in the corners of the pontoon, which serve for hauling purposes; steam winches are also provided for the anchors.

The crane is fitted with three hooks, one each of 140, 20 and 5 tons capacity. The gearings for the various blocks are placed on deck underneath the crane structure. The large winch for the load of 140 tons has a double rope drum and is driven by a steam twin engine of 120-horsepower capacity. This engine also sets the 20-ton winch in motion, as well as the spindles for derricking the crane jib. Both drums of the large winch are built similarly; each one has a diameter of 4 feet 3 inches, a length of 8 feet 8 inches and is made of cast iron. The drums run loosely on axles of 5½ inches diameter. All spur wheels are made of the best steel.

The drums for the 20-ton winch have the same length and are built similarly to the large drums, but with a diameter of only 2 feet 9 inches. A special steam twin engine of 40 horsepower serves for setting the 5-ton winch in motion. This winch has two drums, one to take up the lifting rope and the other one to take up the traveling rope, which sets the trolley, running along the lower flange of the upper part of the jib, in motion.

The following speeds can be obtained:

In lifting a load of 140 tons at the 140-ton hook, about 3 feet 10 inches per minute.

In lifting a load of 70 tons at the 140-ton hook, about 8 feet 2 inches per minute.

In lifting a load of 20 tons at the 20-ton hook, about 26 feet per minute.

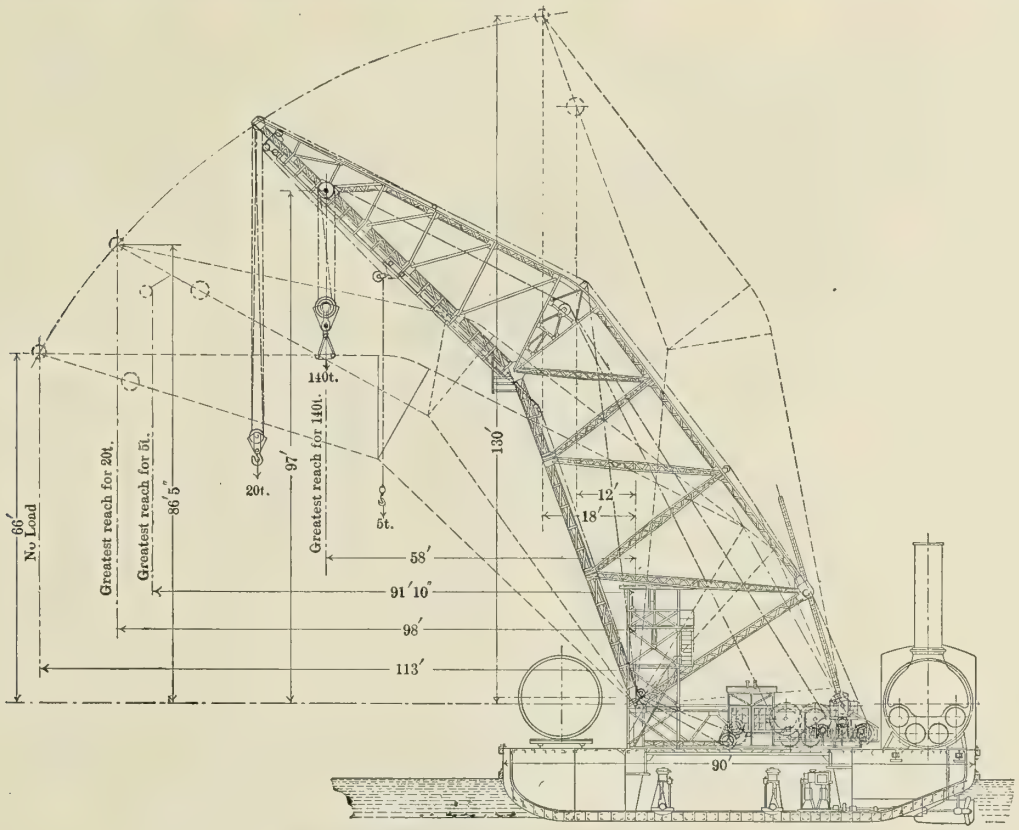


FIG. 4.—SECTIONAL VIEW OF 140-TON FLOATING CRANE.

In lifting a load of 10 tons at the 20-ton hook, about 53 feet per minute.

In lifting a load of 5 tons at the 5-ton hook, about 65 feet per minute.

The crab travels at the rate of about 65 feet per minute.

The general design of the 100-ton crane is about the same as that of the 140-ton structure, but the construction is not so massive. This design has also but two hooks, a main one of 100 tons and an auxiliary one up to 20 tons. The main hook of the 100-ton crane has a maximum outreach of 68 feet 8 inches, and the 20-ton hook, one of 100 feet. With a pontoon whose beam is 66 feet a free space about 24 feet 6 inches remains on deck in front of the jib, in order to ship the articles.

The crane is equipped with a two-cylinder special 9-inch crane engine, rated at about 105 horsepower. This engine drives the four lifting gears and the luffing gear; the large lifting gear is supplied with two drums, each taking one fall of the large lifting rope that carries the 100-ton hook in eight

strands; the smaller hook of 20 tons is suspended by four strands. Besides these at both sides of the framing, two lifting gearings are placed for lifting loads up to 1½ tons.

The working speeds are as follows:

Lifting loads of 100 tons with main hook, about 5 feet 4 inches per minute.

Lifting loads of 40 tons with main hook, about 10 feet 8 inches per minute.

Lifting loads of 20 tons with auxiliary hook, about 28 feet 8 inches per minute.

Lifting loads of 10 tons with auxiliary hook, about 55 feet 6 inches per minute.

Lifting loads of 1½ tons with auxiliary hook, about 90 feet 8 inches per minute.

The luffing of the jib occupies about 12 minutes.

The luffing mechanism operates as follows: At the end of the back stay of the jib two manganese-bronze nuts are inserted, which swing around two cross pinions. The long spindles are screwed into these nuts. The screws are turned with the help of a steam engine placed on deck and corresponding shafting equipped with bevel gears. In consequence the nuts at the ends of the jib, mentioned above, shift on the screws, by which means the jib is drawn in or out, according to the turning direction of the screws.

The steel structure of the crane is composed of light but strong lattice work. At the side of the framing is located the driver's cabin, from which all movements are controlled and in which all necessary hand wheels, brake levers and accessories for controlling the engines are placed.

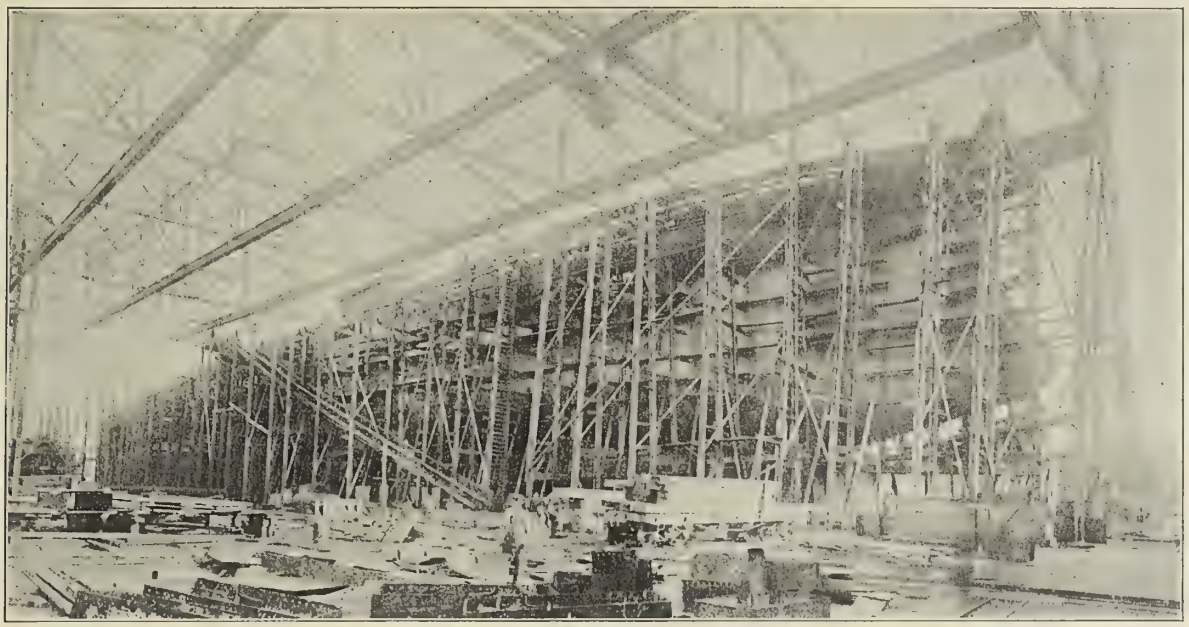
The pontoon has a length of 87 feet 6 inches, a beam of 65 feet 6 inches and a height of 9 feet 10 inches. Two screws are provided for the propulsion of the crane, each being turned by an engine of 60 horsepower. With the above-described pontoon a speed of 3 or 4 knots can be attained. On deck four spills with vertical drums are fitted, which are driven by a special small engine. Opposite the jib a water tank is provided, taking 130 tons and serving as a counterweight. For filling and emptying this tank a steam pump of 1 to 1½ cubic meters per minute capacity is installed. Two boilers are fitted, each having a length of 12 feet 10 inches and a diameter of 6 feet, with a heating surface of about 500 square feet. For ordinary work one boiler is sufficient; the second serves as a reserve and for use in case the pontoon has to travel a long distance. Some cranes of similar design have been built and equipped with gas engines and they have given good service.

THE BATTLESHIP NORTH DAKOTA IMMEDIATELY AFTER LEAVING THE WAYS.

LAUNCH OF THE NORTH DAKOTA.

On Nov. 10 the battleship *North Dakota,* the first of the American *Dreadnoughts,* was launched from the yards of the Fore River Shipbuilding Company, Quincy, Mass. The ceremony of christening the ship was performed by Miss Mary Benton, of Fargo, North Dakota.

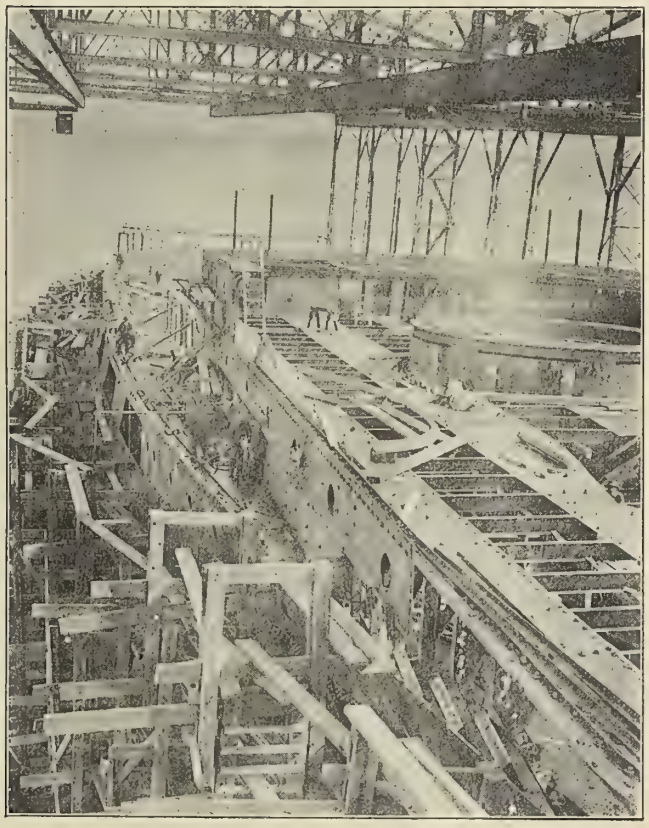
Contracts for the two battleships *North Dakota* and *Delaware,* which were authorized by the Naval Programme of 1906-1907, were awarded Aug. 6, 1907, that for the *North Dakota* being placed with the Fore River Company and that for the *Delaware* with the Newport News Shipbuilding & Dry Dock Co. The keel of the *North Dakota* was laid Dec. 16, 1907, the



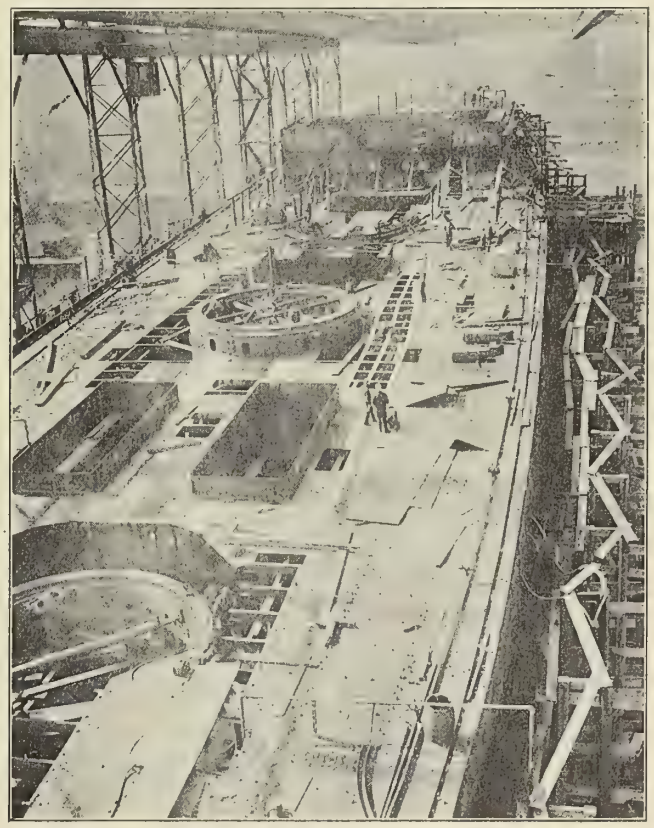
SIDE VIEW OF BATTLESHIP NORTH DAKOTA, OCT. 1, 1908, 54.2 PERCENT COMPLETED.

day on which the Atlantic fleet started on its voyage around the world. The launching was, therefore, accomplished in ten and three-quarters months, or 282 working days after the laying of the keel. This, as a record for speed in warship construction, equals, if not surpasses, the records made in other countries. In one or two instances first-class battleships have been launched in other countries in slightly over eight months, but their percentage completion at the time of launching was not as great as in the case of the *North Dakota*. When the *North Dakota* was launched she was 60 percent completed; and, in addition, much of the vessel's auxiliary machinery, fittings and equipment are already finished and ready for installation, including the five turrets for the main battery. According to the terms of the contract the ship is to be completed July 18, 1910.

The *North Dakota* is 519 feet long over all and 510 feet long on the waterline. She has a beam of 85 feet 2⅝ inches, and a draft at displacement of 20,000 tons of 26 feet 10 inches. At her full-load displacement of 2,275 tons her draft is 27 feet 3 inches. Curtis turbines of a total estimated horsepower of



VIEW OF NORTH DAKOTA, OCT. 1, LOOKING AFT.

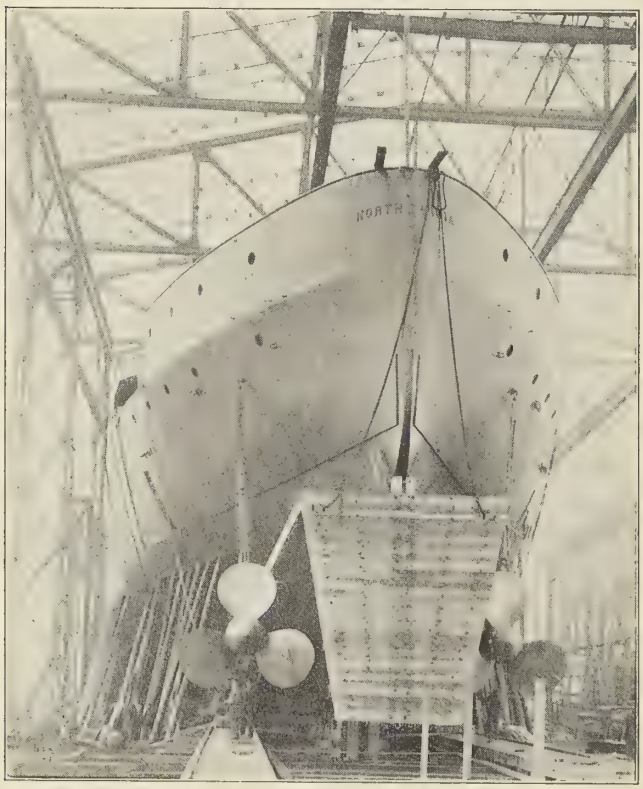


AFTER DECK OF NORTH DAKOTA, LOOKING FORWARD.

25,000 are to drive the ship at a speed of 21 knots, steam being furnished by twelve Babcock & Wilcox watertube boilers, arranged for burning oil in addition to coal. The contract price of the hull and machinery was $4,477,000 (£920,000).

The armament of the *North Dakota* consists of ten 12-inch breechloading rifles mounted in pairs in revolving turrets on the center line of the ship. The arrangement of the turrets is such that all of the 12-inch guns can be fired on either broadside, four of them ahead and four of them astern. To make this possible, two of the turrets are so located that their guns can be fired over the adjacent turret. In addition to the main battery there is a secondary battery of fourteen 5-inch rapid-fire guns for repelling torpedo-boat attack. Ten of these guns are mounted on the gun deck, five on each side, the other four being mounted in sponsons on the gun deck at the bow and stern.

The remaining armament consists of four 3-pounder rapid-fire guns, four 1-pounder semi-automatic guns, two 3-inch



STERN VIEW OF THE NORTH DAKOTA ON THE WAYS.

field pieces, two machine guns of .30 caliber, and two 21-inch submerged torpedo tubes.

The waterline armor belt is 11 inches thick and 8 feet wide, extending 6 feet 9 inches below the full-load waterline. Above this there is a belt 10 inches thick, 7 feet 3 inches wide, and above this, extending to the main deck, the ship is protected by 5-inch armor. The upper plate is 10 feet above the normal waterline and 8 feet or more above the deep-load waterline. The 12-inch turrets are protected by 12-inch and 8-inch armor.

The *Delaware,* which is being built by the Newport News Shipbuilding & Dry-Dock Company, is a sister ship of the *North Dakota,* but she will be driven by two sets of triple-expansion reciprocating engines. Therefore, when both ships have been placed in commission another splendid opportunity will be given to test the relative merits of the old-style reciprocating engine and the leading American type of marine steam turbine. It is expected that the *Delaware* will be launched sometime in February. Both ships are to be fitted with the latest fire-control system, necessitating range-finding positions on top of two lattice-work towers. This type of tower was recently tested on the monitor *Florida,* by placing it under a fire of 6-inch shells in order to demonstrate its stability under attack.

SERVICE TEST OF THE STEAMSHIP HARVARD.*

BY PROF. C. H. PEABODY, W. S. LELAND AND H. A. EVERETT.

The attention of members is particularly called to the fact that this test was run under the conditions obtaining in actual service, no attempt having been made to approximate the ideal conditions existing on the usual trial-trip runs. The engine test began at 9 P. M., when the engine had reached full power, and was stopped at 3 A. M. as we approached Nantucket Shoals. The last reading was taken at 2.55, when the first half-speed bell sounded, and the curves exterpolated to 3 o'clock. The boiler test was begun at 7.10 P. M., and continued until 7 o'clock the next morning, the curves being exterpolated to make 12 hours.

Pressures were read, as far as possible, on the engine-room gages, which were tested for the occasion by the Crosby Steam Gauge & Valve Company. Where engine-room gages were not available, Institute gages, tested at our own laboratories, were used.

The horsepower was determined by means of the Denny & Johnson torsion meter, belonging to the Department of Naval Architecture at the Institute of Technology. The inductors were so set as to give a clear length of shaft between inductors of 63.06 feet on center shaft, 47.52 feet starboard and 49.21 feet port, which gave readings of approximately 95, 75 and 70, respectively, at full power. Inductors were set on all three shafts, but during the test the port recorder failed to work at the end of two hours, due to a rupture of the connections. From the few readings thus obtained, and from readings taken on a preliminary run, when it did work satisfactorily, it appears that the torque readings on this shaft were 93 percent of those on the starboard, and in computing the horsepower this figure has been taken. This, of course, throws some little uncertainty into the calculation, but there is every reason to believe that the assumption is very nearly correct. It would take practically 3 percent error on this shaft to affect the total horsepower by so much as 1 percent. The torsion meter can be read only to single units, so that this of itself means an uncertainty in the last digit, which corresponds to an accuracy of about 1¼ percent. When, however, the readings are plotted at ten-minute intervals, as is here done, and the results faired, the probable error is likely to be much less.

In computing the horsepower 1.506 based on a torsional modulus of elasticity of 11,600,000 was used for the constant K in the formula $H.P. = \frac{K d^4 r R}{C L}$, on which d = the diameter of the shaft in inches (8 inches), r = torsion meter reading, R = revolutions per minute, C = inductor constant (12.5), L = length of shaft in feet between inductors. The shaft was not twisted to determine the exact constant, but since 11,600,000 is the mean of a large number of tests it cannot be far from right. We find 11,000,000 and 12,000,000 to be practically the minimum and maximum values, showing a variation of about 3 percent only in extreme cases. That the actual modulus of the *Harvard's* shaft should vary from the mean value by more than 1½ percent is extremely improbable. All in all, taking into account the uncertainty of the port shaft, there is every reason to believe that the horsepower is subject to an error less than that of the ordinary steam engine indicator.

* Read before the Society of Naval Architects and Marine Engineers, New York, Nov. 20, 1908.

The water consumption was measured by a 6-inch Hersey hot-water meter, loaned by the Hersey Manufacturing Company, of South Boston. It was installed in the suction line between the hot well and the feed pump, and gave exceedingly satisfactory results. This meter was previously calibrated by the Hersey Company, and indicated an under-run of 1 percent under the conditions of the test. The curve of meter readings was plotted without this correction, so that 1 percent should be added to any ordinate of that curve to get the correct reading.

Steam for all the auxiliaries excepting the blowers came through the starboard pipe only, in which a thin plate having a 2½-inch orifice was inserted. The valve on the port line was tightly closed, and the by-pass around the reducing valve on the auxiliary line opened so that all the reduction in pressure was due to the orifice. The reduced pressures were read on the auxiliary steam gage on the engine-room gage board, and the flow of steam computed from coefficients determined by experiments made at our engineering laboratories. An independent orifice was necessary for the blower engines, as this steam did not come from the auxiliary line. The curve of auxiliary steam consumption shows the total from these two orifices.

The probable error in determining the auxiliary steam is small. The gage pressures were read to the nearest pound every ten minutes, and the average must be very nearly right. An error of 1 pound in this average would mean about 2 percent error in the total auxiliary steam, but even if the error here were four times as great it would be of small moment, for the auxiliary steam is only about 13½ percent of the total.

The coal was determined by counting the buckets dumped on the floor. A curve was plotted showing the number of buckets at ten-minute intervals. Every hour the amount of unburned fuel on the floor was estimated, the corresponding points plotted, and the coal-consumption curves drawn through them. The average net weight of coal was 506 pounds per bucket. Such a method as this for coal determination is open to question, but is perhaps the best that can be adopted on a short sea run. Even had the coal been weighed, the different

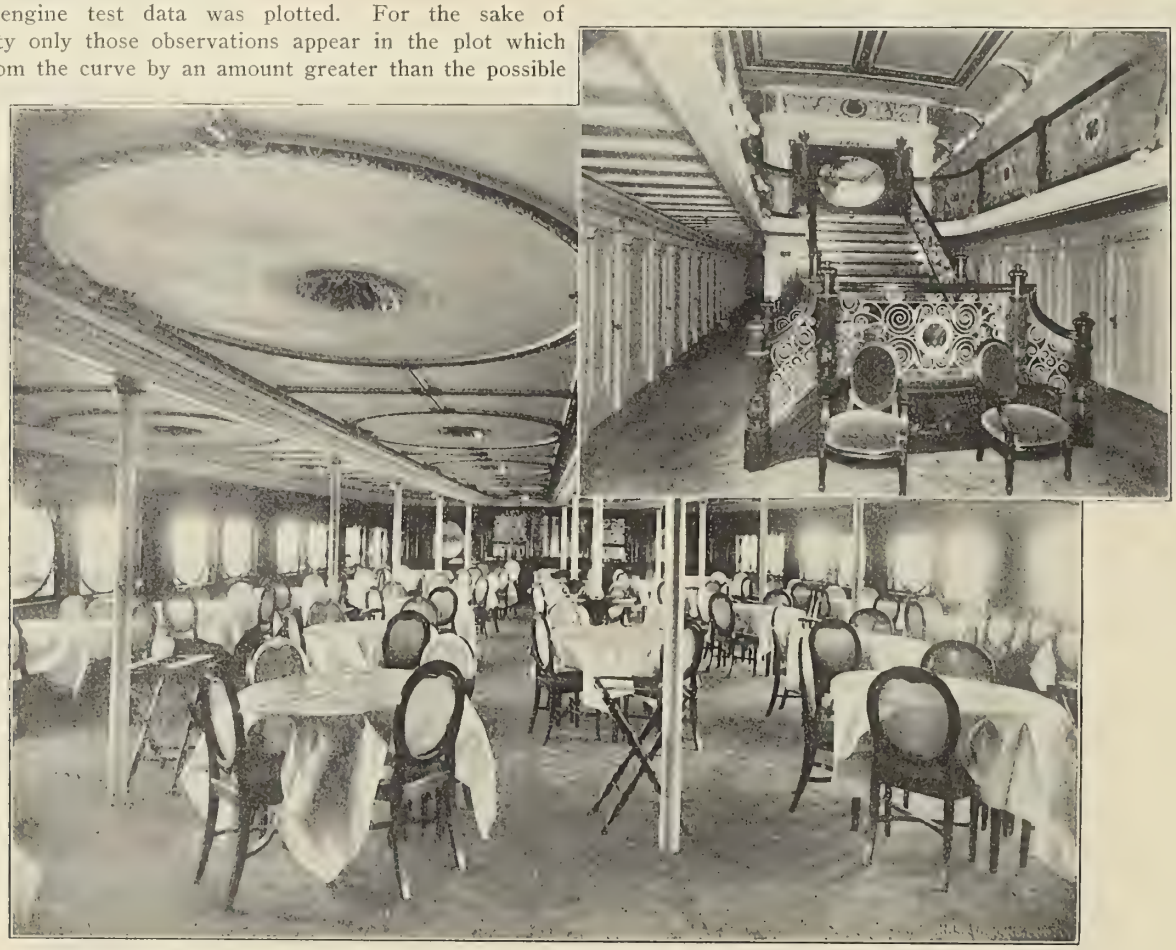
TURBINE STEAMSHIP YALE, SISTER SHIP OF THE HARVARD.

SMOKING ROOM OF THE HARVARD.

condition of the fires at start and finish might have been considerable in so short a run. There is every reason to believe that any determination for so short a run would be somewhat too small. Our figures seem to show 80 tons for twelve hours, equal to 100 tons for fifteen hours, the running time from dock to dock, and this figure checks remarkably well with the amount of coal actually placed on board, an average being taken over a large number of trips.

The quality of steam was determined by the throttling calorimeter, the sample being taken from a tee on the top of the steam main, inserted for this purpose where the pipe was tapped for the gage connection.

The engine test data was plotted. For the sake of simplicity only those observations appear in the plot which vary from the curve by an amount greater than the possible



DINING SALOON AND MAIN STAIRCASE OF THE HARVARD.

accuracy of observation. The close similarity of the curves and the small number of points through which the curves do not pass, is a strong check on the accuracy of the observations.

The following tables give the summary of the results and the principal dimensions of the vessel.

DIMENSIONS.

Length between perpendiculars	386 ft. 6 in.
Breadth of hull molded	50 ft. 6 in.
Breadth over guards	63 ft.
Depth molded	22 ft.
Draft normal	16 ft.
Draft at trial (mean New York to Boston)	16 ft. 2 in.

RESULTS OF BOILER TEST.

Date of test	June 25, 1908.
Duration of test	12 hours.
Boiler pressure average gage	137.8 lbs.
Quality of steam	95.9
Barometer	29.97 in.
Temperature of feed water	203.5° F.
Drfat at blowers (Howden)	1.71 in.
Number of boilers (single-ended Scotch)	12
Total grate area	756 sq. ft.
Total heating surface	29,520 sq. ft.
Ratio heating to grate area	38.6
Total coal fired in 12 hours	179,112 lbs.
Total water fed in 12 hours	2,041,710 lbs.
Equivalent evaporation from and at 212° per lb. of coal	11.2
Coal per sq. ft. grate	19.6

RESULTS OF ENGINE TEST.

Type of engine	Parsons turbine.
Duration of test	6 hours.
High-pressure gage	124.4 lbs.
Low-pressure gage, average S. & P.	21.3 lbs.
Vacuum (average S. & P.)	27.42 in.
Temperature of injection water	63.5° F.
Temperature of discharge water	107° F.
Temperature of hot well	110.8° F.
Total water per hour during test	176,010
Maximum revolutions	p. 468, c. 472, s. 472.
Minimum revolutions	p. 433, c. 444, s. 441.
Average revolutions (6 hours)	455
Maximum shaft horse-power	11,076
Minimum shaft horse-power	9,525
Average shaft horse-power (6 hours)	10,405
Steam for auxiliaries per hour	22,380 lbs.
Steam (4% priming) per shaft horse-power per hour, all purposes	16.9 lbs.
Steam (4% priming) per shaft horse-power per hour, turbines only	14.76 lbs.
B. T. U. per shaft horse-power per minute	265.9
Coal per shaft horse-power (average of 6 hrs.)	1.5 lbs.

A new society, known as the Institute of Metals, recently held its first meeting at Birmingham. This society has been formed for the special investigation of non-ferrous metals and alloys.

Gasoline vs. Alcohol as Fuel for Internal-Combustion Engines.

The technologic branch of the United States Geological Survey, under the direction of Mr. J. A. Holmes, has recently completed an elaborate series of tests on the relative value of gasoline and alcohol as producers of power. The tests, over 2,000 in number, probably represent the most complete and exact investigation of the kind that has been made in any country, and includes much original research work. These tests were conducted at the fuel testing plant of the Geological Survey at Norfolk, Va., under the general charge of Prof. R. H. Fernald and the personal supervision of Mr. R. M. Strong, and show the following results in regard to the comparative fuel consumption of 73 degrees specific gravity gasoline and commercial completely denatured alcohol per unit of power.

Correspondingly well designed alcohol and gasoline engines, when running under the most advantageous conditions for each, will consume equal volumes of the fuel for which they are designed. This statement is based on the results of many tests made under the most favorable practical conditions that could be obtained for the size and type of engines and fuel used. An average of the minimum fuel consumption values thus obtained gives a like figure of eight-tenths (.8) of a pint per hour per brake-horsepower for gasoline and alcohol.

Considering that the heat value of a gallon of the denatured alcohol is only a little over six-tenths (.6) that of a gallon of the gasoline, this result of equal fuel consumption by volume for gasoline and alcohol engines probably represents the best comparative value that can be obtained for alcohol at the present time, as is also indicated by Continental practice. Though the possibility of obtaining this condition in practice here has been thoroughly demonstrated at the Government Fuel-Testing Plant, it yet remains with the engine manufacturers to make the "equal fuel consumption by volume" a commercial basis of comparison.

The gasoline engines that were used in these tests are representative of the standard American stationary engine types, rating at 10 to 15 horsepower, at speeds of from 250 to 300 revolutions per minute, while the alcohol engines were of similar construction and identical in size with the gasoline engines.

The air was not preheated for the above tests on alcohol and gasoline, and the engines were equipped with the ordinary types of constant-level, suction-lift and constant-level-pressure spray carburetors. Many special tests with air preheated to various temperatures up to 250 degrees F., and tests with special carburetors, were made, but no beneficial effects traceable to better carburation were found when the engines were handled under the special test conditions, including constant speed and best load.

The commercial completely denatured alcohol referred to is 100 parts ethyl alcohol, plus 10 parts methyl alcohol, plus one-half of one part benzol, and corresponds very closely to 94 percent by volume, or 91 percent by weight, ethyl alcohol (grain alcohol).

No detrimental effects on the cylinder walls and valves of the engines were found from the use of the above denatured alcohol.

The lowest consumption values were obtained with the highest compression that it was found practical to use; which compression for the denatured alcohol ranged from 150 to 180 pounds per square inch above atmosphere.

Eighty percent alcohol (alcohol and water) for use in engines of the present types would have to sell for at least 15 percent less per gallon than the denatured alcohol, in order to compete with it. The minimum consumption values in gallons per hour per brake-horsepower for 80 percent alcohol is approximately 17.5 percent greater than for the denatured alcohol used or for gasoline. A series of tests made with alcohol of various percentages by volume, ranging from 94 percent to 50 percent, showed that the minimum consumption values in gallons per hour per brake-horsepower increased a little more rapidly than the alcohol decreased in percentage of pure alcohol. That is, the thermal efficiency decreased with the decrease in percentage of pure alcohol. This decrease in thermal efficiency, or increase in consumption, referred to pure alcohol, is, however, comparatively slight, from 100 percent alcohol down to about 80 percent alcohol. Within these limits it may be neglected in making the calculations necessary to compare the minimum consumption values for tests with different percentages of alcohol.

The nearer the alcohol is to pure, the greater the maximum horsepower of the engine. The percent reduction in maximum horsepower for 80 percent alcohol, as compared with that for denatured alcohol used, was less than 1 percent, but the starting and regulating difficulties are appreciably increased.

With suitable compression, mixtures of gasoline and alcohol vapors (double carburetors) gave thermal efficiencies ranging between that for gasoline (maximum 22.2 percent) and that for alcohol (maximum 34.6 percent), but in no case were they higher than that for alcohol. The above thermal efficiencies are calculated from the brake-horsepower and the low calorific value of the fuel, which for the gasoline was 19,100 British thermal units per pound and for the denatured alcohol was 10,500 British thermal units per pound.

As has been previously published, alcohol can be used with more or less satisfaction in stationary and marine gasoline engines, and these gasoline engines will use from one and one-half to twice as much alcohol as gasoline when operating under the same conditions. The possibilities, however, of altering the ordinary gasoline engine as required to obtain the best economies with alcohol are very limited; for the amount that the compression can be raised without entirely redesigning the cylinder head and valve arrangement is ordinarily not sufficient, nor are the gasoline engines usually built heavy enough to stand the maximum explosive pressures, which often reach 600 and 700 pounds per square inch. With the increase in weight for the same sized engine, designed to use alcohol instead of gasoline, comes an increase in maximum horsepower a little over 35 percent, so that its weight per horsepower need not be greater than that of the gasoline engine, and probably will be less.

THE INFLUENCE OF MIDSHIP-SECTION SHAPE UPON THE RESISTANCE OF SHIPS.*

BY D. W. TAYLOR, NAVAL CONSTRUCTOR, U. S. N.

The question of the influence of the shape of the midship section of a vessel upon its resistance is one concerning which the opinions of naval architects differ a good deal. The object of this paper is to lay before the society information throwing some light upon the question, obtained from experiments at the United States Model Basin during the past year. It should be pointed out in advance and carefully borne in mind that, though the actual area of the midship section in a given case may have large influence upon resistance, this is a subject not taken up in the present paper, which deals only with the effect of the shape of the area, not the area itself.

The question of shape of midship section was investigated by means of five types of lines, all the product of Draftsman-in-Charge W. T. Powell. Forty models were tried in all, their data being given in Tables I and II.

Table I refers to the twenty models which had a cylindrical or prismatic or longitudinal coefficient (*l*) of .56, being thus rather fine-ended models. Table II refers to the twenty

* Read before the Society of Naval Architects and Marine Engineers, New York, November 19, 1908.

TABLE I.

Data of models of .56 longitudinal coefficient used for midship section coefficient experiments. Mean immersed length of all models, 20.00 feet. Water-line length of all models, 20.512 feet.

Model No.	Displacement of Model in Fresh Water, Pounds	Midship-section Coefficient	Model Dimensions. Beam, Feet.	Model Dimensions. Draft, Feet.*	Displacement length Coefficient, Salt Water.
826	....	.70	4.187	1.432	
830	....	.80	3.916	1.340	
834	3,000	.90	3.692	1.263	159.6
838	....	1.00	3.503	1.198	
842	....	1.10	3.339	1.143	
827	....	.70	3.627	1.241	
831	....	.80	3.393	1.161	
835	2,250	.90	3.199	1.094	119.7
839	....	1.00	3.035	1.038	
843	....	1.10	2.893	.990	
828	....	.70	2.961	1.013	
832	....	.80	2.769	.948	
836	1,500	.90	2.611	.893	79.8
840	....	1.00	2.477	.847	
844	....	1.10	2.361	.808	
829	....	.70	2.417	.827	
833	....	.80	2.261	.774	
837	1,000	.90	2.132	.729	53.2
841	....	1.00	2.022	.692	
845	....	1.10	1.928	.660	

*All models tested on even keel.

TABLE II.

Data of models of .68 longitudinal coefficient used for midship-section coefficient experiments. Mean immersed length of all models, 20.00 feet. Water-line length of all models, 20.512 feet.

Model No.	Displacement of Model in Fresh Water, Pounds.	Midship-section Coefficient.	Model Dimensions. Beam, Feet.	Model Dimensions. Draft, Feet.*	Displacement length Coefficient, Salt Water.
889	....	.70	3.283	1.123	
893	....	.80	3.071	1.051	
897	2,250	.90	2.895	.991	119.7
901	....	1.00	2 747	.940	
905	....	1.10	2.618	.896	
890	....	.70	2.681	.917	
894	....	.80	2.508	.858	
898	1,500	.90	2.364	.809	79.8
902	....	1.00	2.243	.767	
906	....	1.10	2.138	.732	
891	....	.70	2.188	.749	
895	....	.80	2.047	.701	53.2
899	1,000	.90	1.930	.660	
903	....	1.00	1.831	.626	
907	....	1.10	1.745	.597	
892	....	.70	1.548	.530	
896	....	.80	1.448	.495	
900	500	.90	1.365	.467	26.6
904	....	1.00	1.295	.443	
908	....	1.10	1.234	.423	

*All models tested on even keel.

models which had a longitudinal coefficient of .68, being thus full-ended models. The ratio of beam to draft for all models was 2.923.

It will be observed that the slenderness of these models, if I may apply this expression, is characterized by what I call "Displacement Length Coefficient," or $\frac{D}{\left(\frac{L}{100}\right)^3}$, where D is displacement in salt water in tons and L is length on water line in feet. This coefficient is also the displacement in tons of a vessel upon the lines of the model and 100 feet long. It is a convenient coefficient to use when dealing with questions of speed.

There were five separate midship-section coefficients (m) used; namely, .7, .8, .9, 1.0 and 1.1. Each coefficient was used with the two longitudinal coefficients and with four displacements or four displacement length coefficients. Fig. 1 shows the five types of lines for five vessels 400 feet long and of 7,662 tons displacement, the longitudinal coefficient of each being .56. Change of displacement for fixed longitudinal and midship-section coefficients was obtained by changing both beam and draft the same percentage. The same parent lines were used for each longitudinal coefficient.

The variations in fullness were obtained by shifting the sections fore-and-aft. Fig. 2 shows the two curves of sectional area used for the longitudinal coefficients (l) .56 and .68. They both, in this case, refer to a 400-foot vessel of 7,662 tons displacement. For identical displacements as in Fig. 2, the actual midship-section area for the .68 coefficient is less than that for the .56 coefficient, but the midship sections themselves are similar. The similar area corresponding to the .68 coefficient was obtained by reduction of dimensions from the larger area required by the .56 coefficient. The method of passing from one form to another is indicated in the figure. Thus at station 33 the area required for the .68 coefficient vessel is 582 square feet. This area is 582 ÷ 987 or .590 of the midship area of the .68 coefficient vessel, which is 987 square feet. The midship area of the .56 coefficient vessel is 1,198 square feet; .590 of this 707 square feet, which is found at station 30, hence the section at 33 on the .68 ship is the section at 30 on the .56 ship reduced in size from 707 square feet to 582 square feet, both beam and draft being reduced in the same proportion.

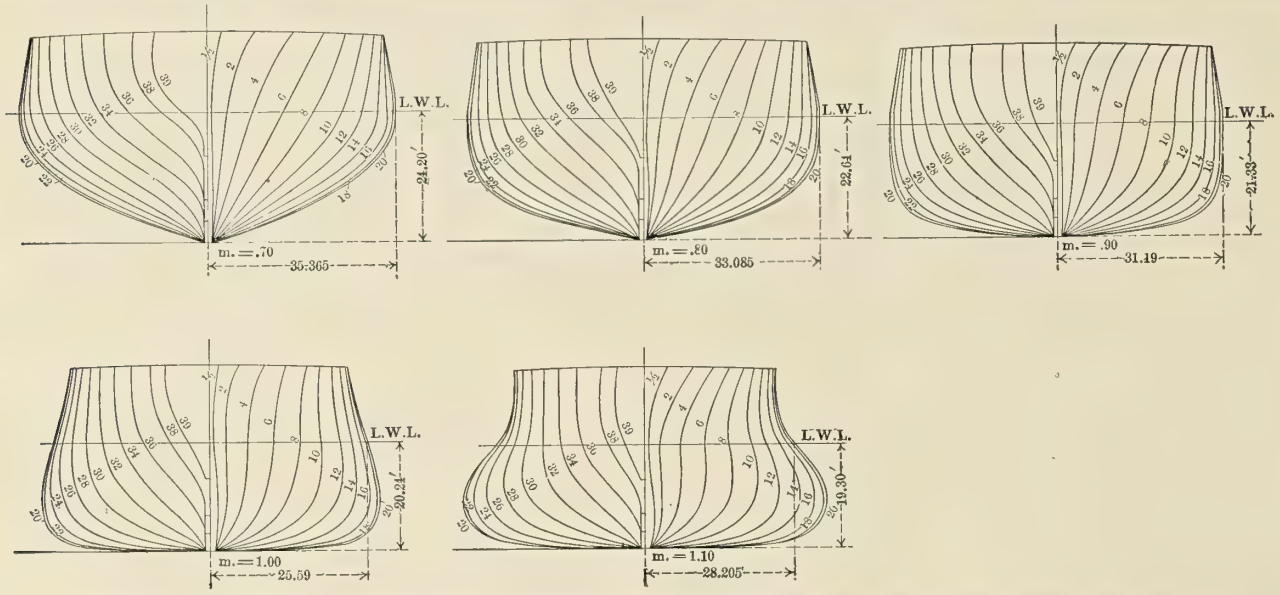


FIG. 1.—BODY PLANS FOR 400-FOOT, 7,662-TON VESSELS OF FIVE DIFFERENT MIDSHIP SECTION COEFFICIENTS. LONGITUDINAL COEFFICIENT .56.

Taking up now the relative resistances, these may be considered under two heads; namely, frictional resistance and residuary resistance, the latter being mostly wave making. The frictional resistance is assumed to depend only upon the wetted surface. The wetted surface is very conveniently ex-

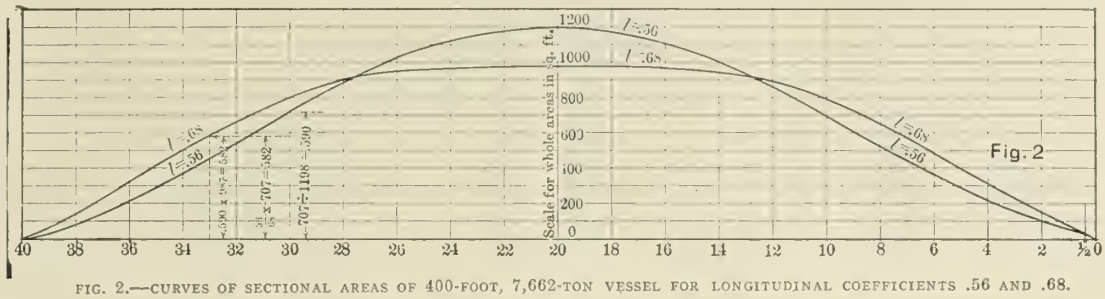


FIG. 2.—CURVES OF SECTIONAL AREAS OF 400-FOOT, 7,662-TON VESSEL FOR LONGITUDINAL COEFFICIENTS .56 AND .68.

pressed by a formula in a paper by me to be found in Volume I (1903) of the transactions of the society; namely, $S = C\sqrt{DL}$ where S is the wetted surface in square feet, C is the coefficient, D is the displacement in tons in salt water, and L is the mean immersed length in feet.

Figures 3 and 4 give curves of wetted surface coefficient deduced from calculations for the forty models and plotted upon midship-section coefficient in each case. Fig. 3 refers to the longitudinal coefficient of .56 and Fig. 4 to the longitudinal coefficient of .68. The four curves of each figure refer to separate values of displacement length coefficient as indicated.

It is seen that, broadly speaking, there is in each case a minimum wetted surface which occurs for midship-section coefficient a little above .90 in the case of the fine-ended models, and a little below .90 in the case of the full-ended

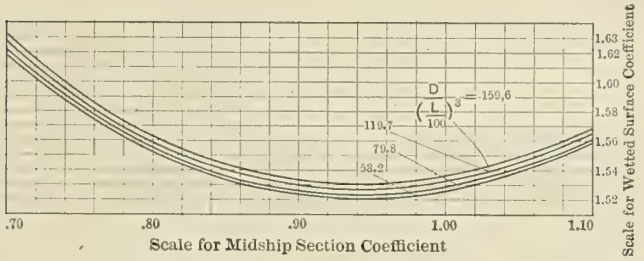


FIG. 3.—CURVES OF WETTED-SURFACE COEFFICENTS. LONGITUDINAL COEFFICENT .56.

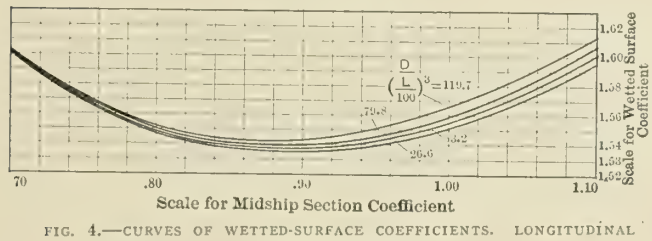


FIG. 4.—CURVES OF WETTED-SURFACE COEFFICIENTS. LONGITUDINAL COEFFICIENT .68.

models. Very fine midship section and very full midship section both involve an increase of wetted surface, but since, in practice, midship-section coefficients greater than unity are not used, the practical conclusion to be drawn from Figs. 3 and 4 is that for midship-section coefficients between .85 and 1.00 there is little variation of wetted surface due to the midship-section coefficient alone. For very fine midship sections, as .7 for example, there is an appreciable increase in wetted surface and a corresponding increase in frictional resistance, although this increase is not serious. In passing, it may be remarked that a comparison of Figs. 3 and 4 will show that for midship-section coefficient about .84 the absolute wetted surfaces are almost identical both for the fine-ended ships and the full-ended ships. Above this coefficient the wetted surfaces are slightly less for the fine-ended ships, and below this coefficient they are slightly less for the full-ended ships.

Coming now to the question of residuary resistance, Figs. 5 and 6 give forty curves of residuary resistance obtained from the forty models tried. These curves are plotted not on actual speed or in terms of actual resistance, but upon the speed-length ratio $\frac{V}{\sqrt{L}}$ and with resistance expressed in pounds per ton. This means that they are applicable, regardless of size. It will be seen that, broadly speaking, within the limits of practicable speed, the residuary resistance is less the greater the midship-section coefficient.

For Fig. 6 (longitudinal coefficient .68) the residuary resistance is less at all speeds, with almost no exception. For

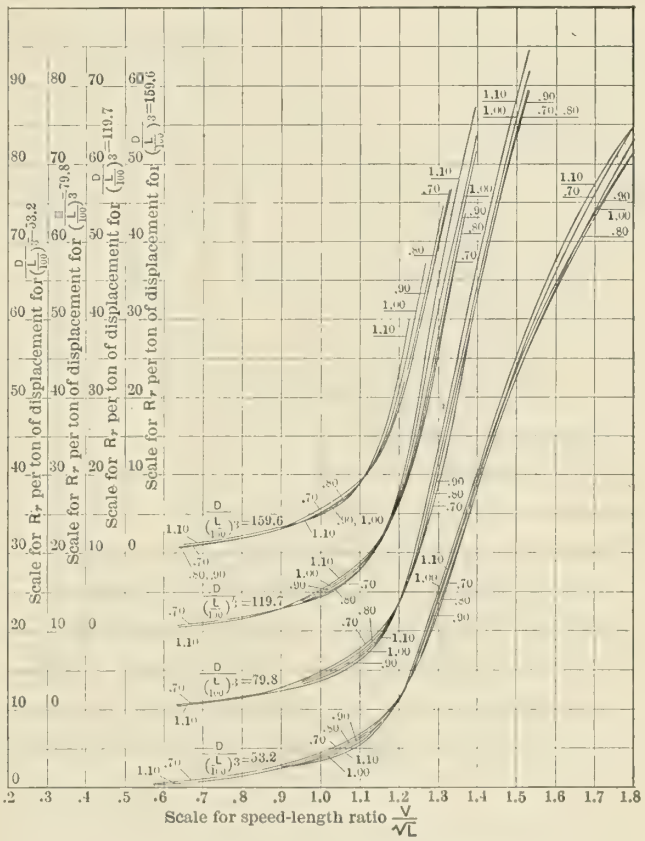

Scale for R_r per ton of displacement for D/(L/100)^3 = 53.2
Scale for R_r per ton of displacement for D/(L/100)^3 = 79.8
Scale for R_r per ton of displacement for D/(L/100)^3 = 119.7
Scale for R_r per ton of displacement for D/(L/100)^3 = 159.6
.2 .3 .4 .5 .6 .7 .8 .9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7 1.8
Scale for speed-length ratio V/√L

FIG. 5.—CURVES OF RESIDUARY RESISTANCE PER TON OF DISPLACEMENT. LONGITUDINAL COEFFICIENT .56.

the curves of Fig. 5 (longitudinal coefficient .56), at speeds below the speed-length ratio of from 1.1 to 1.2 the large midship-section coefficients have the advantage. Above this point in Fig. 5 the results are somewhat confused, but, as a general thing, the finer the midship-section coefficient the less the resistance. It should be borne in mind, however, that these

higher speeds are beyond the speeds obtained in practice by vessels of this type. For such high speeds the longitudinal coefficient of .56 is too small and, while .68 is rather high, it is closer to the best coefficient than .56. It will be found, for instance, that at the highest speeds the resistances for the same values of displacement length coefficient are less in Fig. 6 than in Fig. 5.

Figs. 3, 4, 5 and 6 deal with separate elements of resistance. As expressing more definitely concrete results, curves of total effective horse-power as estimated from the experiments, for a large vessel of .56 longitudinal coefficient and for a small vessel of .68 longitudinal coefficient were plotted. Five curves corresponding to the five midship-section coefficients were made in each case, and it is seen that here, too, the large coefficients have a little the best of it.

A natural tendency is to associate fineness with speed and to suppose that a fine midship section is favorable to speed.

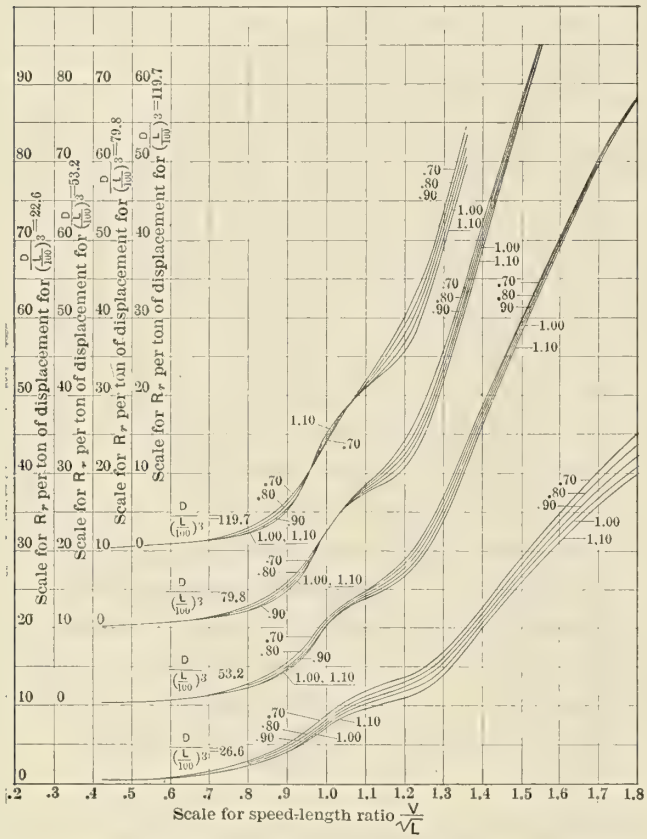


FIG. 6.—CURVES OF RESIDUARY RESISTANCE PER TON OF DISPLACEMENT. LONGITUDINAL COEFFICIENT .68.

The results of experiments with the forty models laid before you in this paper show that, as regards these models, fineness of midship section is by no means favorable to speed, and furnish a presumption that, for vessels generally of ordinary types, it is not favorable to speed. Of course it cannot be claimed that the forty models tried cover the whole possible field. For instance, I hope to have an opportunity to make some experiments on this subject where, instead of varying both beam and draft to obtain variation of midship-section coefficient, draft is varied and beam not changed. It seems reasonable to conclude, however, that the shape of the midship section has a comparatively minor influence upon the speed, and that the midship-section coefficient may range from .85 to 1, with very little effect upon speed, and may be made as low as .70 without a material increase in resistance.

A reasonable explanation of the results I have laid before you may, I think, be deduced from an inspection of Fig. 1. It is seen that the fine midship sections are associated with the large dimensions. This is necessarily the case when we separate the question of midship-section coefficient from the question of midship-section area. If area is constant, smaller coefficients must mean larger dimensions. For instance, keeping the area of midship section constant for the vessels of Fig. 1 and passing from the extreme midship-section coefficient of 1.1 to .7, we increase the draft from 19.3 to 24.2 feet and the beam from 56.41 feet to 70.73 feet. Increased beam and draft mean increased disturbance of at least a portion of the water through which a ship passes.

The results of the special series of experiments on this subject are in full accord with our general experience at the Model Basin. Among the very large number of model experiments we have made, we have had no results indicating any material influence of shape of midship section upon resistance. For vessels of extreme types and extraordinary speeds there may be great virtue in some special form of midship section—yet to be discovered—but our general model-basin experience and the special experiments described above appear to warrant the conclusion that for vessels of usual types and of speeds in knots no greater than twice the square root of the length in feet, the naval architect may vary widely midship-section fullness without material beneficial or prejudicial effect upon speed.

STEAMSHIP ENGINEERING ECONOMIES.

BY A MARINE ENGINEER.

In these days of fierce and, in many cases, foolish competition, no occupation feels the effect of this stress more than the shipping industry, especially as it applies to our seaborne freight-carrying trade. It therefore behooves all those interested in the country's welfare, and particularly those concerned in the shipping trade, to put forward every effort to keep working costs down to a minimum.

Taking, for example, the case of a steamer after it has left the home port, the shipowner or his representatives practically lose all control over the vessel, and must of necessity leave it entirely in the hands of the ship's crew, from the captain downwards. While admitting that the captain of a ship can save a good deal of money to his owners by taking the shortest and safest course between two ports, and also by seeing to it that he gets as quick a discharge for his cargo as possible when in port, it must also be allowed that a good deal of the economical running of a ship depends upon the care and efficiency with which the engineering staff of the boat carry out their duties. It may, therefore, be worth while going over a few of the points in which engineering economies may be made in marine practice.

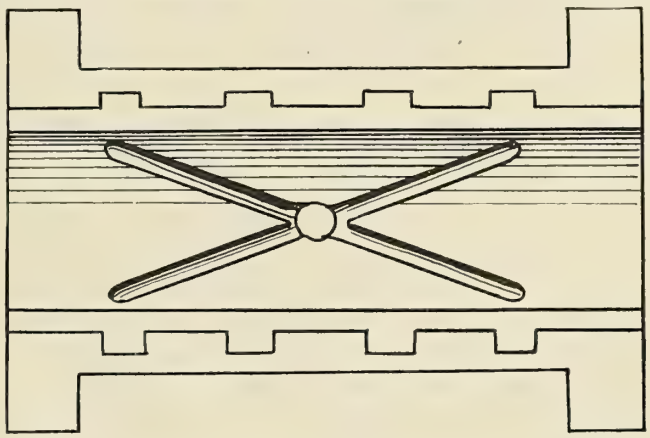
Economies in the steamship, as in other branches of engineering operations, begin with the boiler plant, and unless this part of the ship has the utmost care of the staff a great deal of expensive steaming will be the result. In the first place, the boilers ought to be kept as free from scale and dirt as possible. This can be done effectually and well by periodically cleaning them externally and internally, and by the use of an evaporator for treating the make-up water required for the boilers. The use of a feed-water heater would also be of great advantage in tending to cause the water to deposit any foreign matter in the heater where it can be easily blown off instead of letting it get into the boiler, where a complicated cleaning process is necessary. A certain kind of efficient boiler compound is also very useful in removing scale from boiler surfaces and in the prevention of corrosion. As, however, the writer holds no brief from the makers of this compound its name need not be stated, although he has used it with advantage in a number of boilers where scale and corrosion were very troublesome. Another simple and effective method of preventing corrosion is to apply a wash of Portland cement to the boiler plates. This prevents any acid compound in the feed water from attacking the iron of the plates.

A saving in coal consumption can be made by the use of a superheater. It is well known that a saving up to 10 percent can be effected by this means, and in the writer's opinion nothing much can be gained by using steam in a reciprocating engine at a higher temperature than 450 degrees. By placing the superheater at the root of the funnel and allowing the escaping gases from the boiler to pass round it, a sufficiently large saving is the result to warrant the capital expenditure on the superheater to be made. It should hardly be worth mentioning, except for the lamentable failures sometimes found on board ship, that all boiler connections should be kept in as good condition as possible, especially the blow-down and scum cocks.

The fire should be very carefully attended to. One of the chief points of supervision should be to make sure that the fireman takes as short a time as possible in cleaning fires. He should also be instructed to keep as even and moderately thick a fire as possible. In a natural draft stokehold, for example, fires from 4 inches to 4½ inches are usually the best practice. It is a common occurrence for firemen to fill up their fires as much as they will hold, in order to enable them to get a longer idle spell, and it is extremely difficult to get this class of man to desist from this bad habit. Probably, however, a little encouragement in the form of a bonus given to the men who keep the best fires will go a long way towards leading to more economical coal consumption. The firemen should also be very carefully taught the use of the dampers, to allow the proper amount of air to pass into the furnaces, increasing the percentage of carbon dioxide in the furnace gases. Assuming that the boilers were in the first place well designed, the efficiency maintained in the stokehold of a ship would be very high if the foregoing points were carefully attended to.

We now come to the consideration of the engine room, the first point of interest being the main engines. The piston, packing rings and valves of these engines ought to be maintained in as good a condition as possible. A very fruitful source of trouble and expense is found in the heating of the bearings on the crank pins and also in the main bearings, thus causing a stoppage or slowing down of the engines. This trouble could be almost entirely obviated, assuming that the bearings had been designed with liberal bearing surface, either by a more careful regulation of the oil supply or by the provision in the bearings of properly constructed oil channels. In the writer's experience with two sets of quadruple expansion engines of very large horsepower, a great deal of trouble and stoppage was caused on the first voyage by the heating of the crank pin bearings, notwithstanding the fact that the engines were built by a firm of high-class engine builders. On arrival at the home port all of the bearings were taken apart and thoroughly examined. The white metal was found to be of good quality, and oil channels were provided in the bearing metal. It was, however, considered that these oil channels were not all that could be desired, and they were further altered by being cut, as shown in the sketch, the channels being made ⅛ inch deep by ¼ inch broad, and well rounded at the edges. This was done to both the crown and bottom brasses, and, apparently, this cured the trouble, because the bearings worked perfectly cool on the next trip.

In marine work the oiling system as arranged for the usual reciprocating engine is far from being perfect. In most cases it consists simply of a box filled with oil fitted to each bearing, the lubricating action being dependent on the capilliary effect provided by the siphons. A much better arrangement would be to have the oil tank placed above the top grating in the engine room, and fastened to the bulkhead; the bearings would then be fed with oil at a pressure corresponding to the height between the tank and the engine bearings. The crank pits of the different engines could be semi-enclosed, which would prevent loss of oil, and made watertight in order to prevent the bilge water from mixing with the oil; such oil and water as collected in the crank pits could be led by pipes into a tank made in the form of a combined cooling tank and filter, placed in the same position as the engine room ballast tank. This tank could be fitted with two fine gauze screens, of different mesh, to catch any impurities in the oil, and the lubricant could then be pumped through two or three coils of tubes on its way back to the storage tank at the top of the engine room. A portion of the circulating water for the condenser could be used for the purpose of cooling the oil, and the cost would not be much greater than that of the systems in vogue at the present time. In any case, it would more than repay itself for any extra outlay in the smoother running of the engine and decreased number of hot bearings. One greaser would be able to look after a larger amount of the plant, as he would have little or nothing to do with the exception of swabbing the rods. In large ships carrying more than one greaser per watch a saving of a greaser's wages would be found possible.



SKETCH SHOWING ALTERATION OF OIL CHANNELS IN CRANK-PIN BEARINGS.

Another source of economy would be found in the use of metallic packing on the piston rods of all the engines, as this class of packing requires very little lubrication and adjusts itself very readily to the working position of the piston rod. In the writer's experience the rod has been out of line to the extent of 3/16 inch, and yet the packing has worked perfectly satisfactorily. Moreover, the up-keep of this type of packing is very small. The writer has known it to run for four years without being looked at, and at the end of that time the only refitting necessary was that one of the rings had to be refilled with white metal. After an experience of ten years with this packing the writer may say that he has not once had to stop an engine owing to trouble with the packing after it had been properly fitted and adjusted. It may possibly leak a little for a day or two after being fitted to the gland until it becomes bedded on the rod, but after this it will remain perfectly dry.

In large vessels the auxiliary plant forms a very considerable part of the total horsepower of the ship and requires a considerable amount of attention in up-keep. In the case of reciprocating engines, where the air and circulating pumps are driven from the intermediate pressure engines, this is found to be a most economical arrangement, but in the case of large ships the circulating pump is usually driven by an independent engine. A much better method of driving this would be by means of an electric motor, and where the bilge pumps are driven by a separate engine a motor would also be an advantage in place of the steam engine. Deck pumps and smaller auxiliary apparatus should also be run electrically, and this could be easily and cheaply arranged, as most of the modern ships have an electric plant on board for lighting. This plant could be increased in size for comparatively small capital cost to enable it to take the load of these extra motors. In most ships the electrical generating plant has to run for 24

hours per day, with often an output of not more than 10 percent of the full load of the machine. By having the pump load in addition to the small day lighting load, the output of the machine could be increased to something like its full load, which would increase the efficiency of the machine and reduce the running costs very considerably. Moreover, as a stand-by set is usually provided this could be run for peak loads in the evening, and this would not only increase the economy of the engine room in steam consumption, oil, repairs and attendance, but space and weight would also be reduced, giving a greater margin for the carrying capacity of the ship.

It will now be interesting to consider the deck plant, such as the capstan, steering engine and engines in the case of winches driven by steam. There is, first of all, the long length of steam pipes which have to be carried from one end of the ship to the other, with valves scattered here and there, and the consequent loss due to condensation and radiation of heat. In addition to this there is the leaky joint, which is almost inseparable from this class of work and which deteriorates the wood planks composing the decks very rapidly. Again, the uneconomic winch engine itself very frequently has its cylinder quite uncovered with the exception of a coat of paint. It usually boasts of a leaky stop valve and slide valves; there is general heavy wear and tear on the brasses of this very useful but uneconomical machine, and lastly, but by no means least, there is the hideous noise made by its gearing. By having these machines driven by electric motors, no unsightly and leaky steampipes would be seen, wear and tear would be materially reduced, and noise, with its consequent misunderstandings between the men in the hold and those on deck, would be abolished; and also the saving in steam would be very large. Supposing, however, that a motor drive cannot be applied to a ship which has a steam winch, it should at least be arranged so that the exhaust steam of the winch engine would be utilized to heat the feed water of the main boilers by means of an exhaust feed heater. The steering engine would be replaced by a motor working with a reduction gear, and the capstan could be dealt with in precisely the same way. When ash expellers are not installed a motor could be installed for hoisting up the ash from the stokeholds.

In a number of the modern liners some of the improvements suggested above have, of course, been adopted, but the majority of ships still have a long way to go in adopting more economical methods of driving their auxiliary plants. Probably this conservatism is largely due to the fact that the consulting engineers of these ships have had little or no experience with electrical machinery and are afraid to recommend its use in vessels under their supervision. The idea seems to have become fixed in their minds that electric motors are not so reliable as a steam plant, but this feeling ought to be driven from their minds by inviting them to investigate for themselves the amount of electrically-driven auxiliary apparatus which is used in all our large electric generating stations throughout the country. Moreover, it should be remembered that there are numberless large manufacturing industries which have taken up the electrical system of driving, and which run continuously day and night without any trouble. One has only to look at the progress which is being made every day in the electric driving of fans and pumps in large collieries throughout the country to understand how reliable the electric plant has become, inasmuch as many hundreds of men are dependent upon the uninterrupted operation of these essential parts of a mine equipment. Manufacturers' representatives do not push this part of the field for electric plant as keenly as they might do, and a good deal of work is waiting for the firms who can successfully take up the work of equipping our mercantile marine.

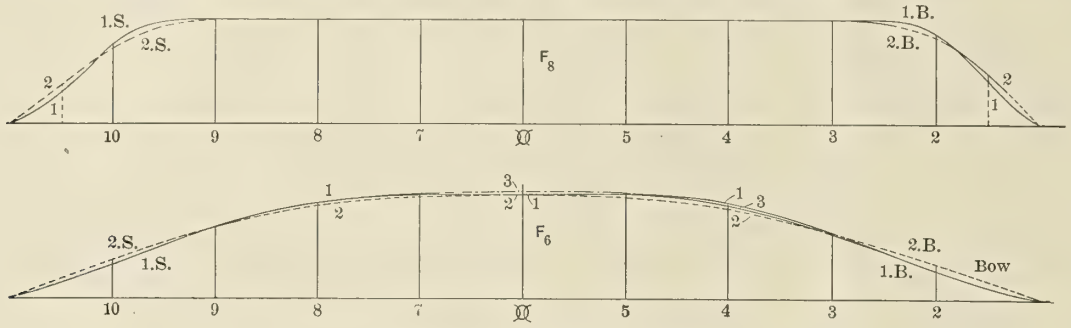


FIG. 1.—CURVES OF SECTIONAL AREAS.

FURTHER EXPERIMENTS UPON LONGITUDINAL DISTRIBUTION OF DISPLACEMENT AND ITS EFFECT UPON RESISTANCE.*

BY PROFESSOR HERBERT C. SADLER.

In a paper read before the society last year the results of some experiments upon resistance as affected by distribution of displacement were given for one form of .73 block coefficient. Since that time similar experiments have been carried out upon the same general lines, for finer and fuller forms, the particulars of which, including those of last year (F7), are given below.

PARTICULARS OF MODELS.

Series No.	$\frac{L}{B}$	$\frac{B}{d}$	$\frac{L}{d}$	Coefficients.		
				Block.	Prism.	Midship.
F6 (1)	8	2.142	17.142	.6533	.6778	.9638
F7	8	2.142	17.142	.733	.760	.964
F8	8	2.142	17.142	.855	.869	.984

The curves of sectional areas are shown in Fig. 1. In each case the same general idea has been followed, viz., the dimensions, displacement and coefficients have been kept constant for each series and the distribution of displacement modified by altering the curve of sectional areas. In the models where a parallel middle body was used, this was held rigidly for the distance shown on the curve of sectional areas, and was therefore "actual parallel middle body" and not "virtual middle body." The general shape of the sections was also kept constant. The various models were tried at three different drafts and some at different trims, but those for the maximum draft only are given, as the resistance curves for the other drafts follow the same general form.

The fine bow and stern accompanied by parallel middle body are marked 1B and 1S, and the fuller bow and stern with no middle body, or, in the case of the fullest type, a reduced middle body 2B.2S. Combinations of the two, such as the fine bow with the full stern, are distinguished by the letters 1B.2S. In each case four models were made representing all the combinations possible with the given curves of sectional areas.

* Read before the Society of Naval Architects and Marine Engineers, New York, November 19, 1908.

Dealing in the first place with the fullest form F8, it will be noticed that there is not much scope for very great variation in form owing to the shortness of the ends. In the models tried, the difference between the two extreme forms is represented by an increase of 10 percent in length of middle body. The curves of residuary resistance per ton of

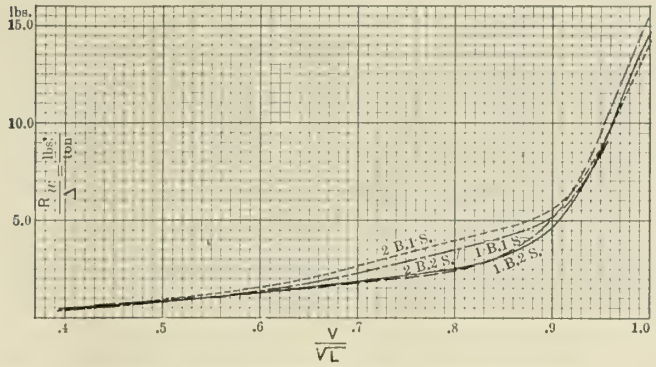


FIG. 2.—CURVES OF RESIDUARY RESISTANCE PER TON OF DISPLACEMENT FOR MODEL F6.

displacement are shown in Fig. 3 for the various combinations of bows and sterns, and the comparative results in the following table:

COMPARATIVE RESIDUARY RESISTANCE.

$\frac{V}{\sqrt{L}}$	1B.1S.	1B.2S.	2B.1S.	2B.2S.
.5	100	80	84.5	56
.55	100	77.7	79.0	50.6
.6	100	74	72	53.0

The form with the fine ends and longer middle body is decidedly the worst, while that with the shorter middle body and fuller ends is the best. Between the combinations of ends there is little to choose. The results for this form are therefore the opposite to those obtained for the finer forms, but the explanation of this is probably as follows. Owing to the shortness of the ends it is possible to obtain only a very short length of line at the extreme end, which can be reduced in fullness. This reduction must be immediately followed by an increase and, consequently, a rather abrupt shoulder where the lines run into the middle body. In fact, in the form with

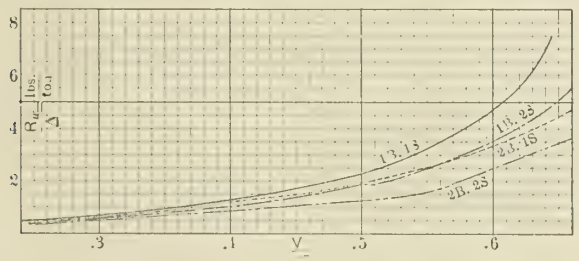


FIG. 3.—CURVES OF RESIDUARY RESISTANCE PER TON OF DISPLACEMENT FOR MODEL F8.

the finer ends, the length could be reduced about 2 percent at each end by "snubbing," with little or no effect upon the resistance. It will also be noticed that although the angle of entrance at the extreme end is much less in the form with the fine ends, the real mean angle of entrance over, say, one-half of the fore or after body, is actually less in the form with the fuller ends.

Reference to the wave profiles for the different forms shows what happens at the ends, and explains some of the difference between the resistance curves for 1B.1S and 2B.2S. At almost any speed there is a marked secondary bow wave in the forms with the fine ends, at the point where the fore body begins to run into the middle body. The two forms of stern show somewhat the same effect, a marked hollow occurring in the fine stern type, in about the same relative position from the stern.

Referring now to the finer form F6, the curves of sectional areas marked 1B.1S and 2B.2S represent the various modifications in the different models; the form with the fine end having 20 percent, and that with the full ends no middle body. The curves of residuary resistance are shown in Fig. 2, and the comparative results in the following table:

COMPARATIVE RESIDUARY RESISTANCES.

$\frac{V}{\sqrt{L}}$	2B.1S.	2B.2S.	1B.1S.	1B.2S.
.75	100	86.5	62.6	64.5
.8	100	87.6	61.7	63.5
.85	100	91.2	71.7	69.5

In this case the form with the full bow shows a marked inferiority to that with the fine bow, within the range of speed suitable for this form. At higher speed-length ratios the somewhat easier form represented by 2B.2S seems to show up a little better than the one with the finer bow. The fuller and easier after-body also seems better than the finer form. On the same basis of curve of sectional areas in the above series, two more models were made, but of greater

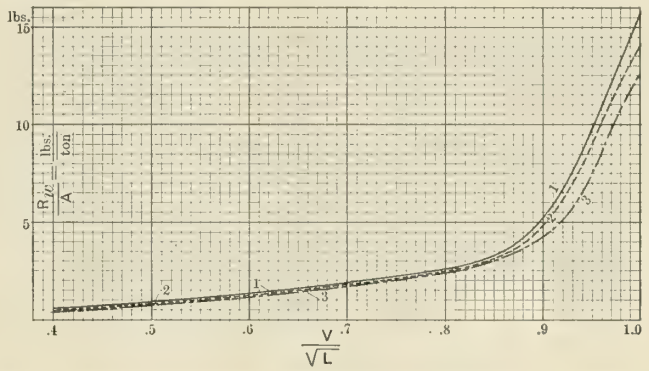


FIG. 4.—CURVES OF RESIDUARY RESISTANCE FOR MODELS F6 (1, 2 AND 3).

beam. No. 2 had exactly the same curve of sectional areas as No. 1, the beam only changed. In No. 3 the same beam was taken as No. 2, but the area of the midship section was increased to compensate for the reduction of displacement, due to cutting away the form between sections 3 and 5 (Fig. 1). Nos. 1 and 2 have therefore the same prismatic, but different block coefficients, while Nos. 2 and 3 have the same block but different prismatic coefficients. The lines at the extreme ends in all cases are the same.

TABLE OF PARTICULARS.

No.	$\frac{L}{B}$	$\frac{B}{d}$	$\frac{L}{d}$	Block.	Prism.	Midship.
F6.1	8	2.142	17.142	.6533	.6778	.9638
F6.2	7.272	2.142	17.142	.594	.6778	.874
F6.3	7.272	2.142	17.142	.594	.664	.895

The curves of residuary resistance are shown in Fig. 4, and the comparative results in the following table:

COMPARATIVE RESIDUARY RESISTANCES.

$\frac{V}{\sqrt{L}}$	F6.1	F6.2	F6.3
.8	100	96	96
.85	100	95.4	92.3
.9	100	92.3	81.2

Up to a speed-length ratio of .8 there is very little to choose between the three forms, but above that speed the effect of widening and fining the midship section is apparent by comparing curves 1 and 2. By reducing the prismatic coefficient

still further, as in No. 3, there is a further reduction in residuary resistance. In fact the curve of sectional areas of No. 3 has all the good and none of the bad points of Nos. 1 and 2, the ends being fine and the lines towards the middle of a gradual character with no sudden change.

In practice, especially where the mold system is used, it is of advantage from a builder's point of view to have as much parallel middle body as possible. In the above series and in those given last year, this point has been kept in view. The three series, F6, F7, and F8, represent a range of models of .6 to .85 block coefficients.

An investigation of the curves of residuary resistance shows that in the first place it is of decided advantage to use a parallel middle body and a fine bow up to forms with about .8 block coefficient. Beyond that degree of fullness, the reverse is true, owing to the reasons discussed above.

There is no doubt that further improvements in performance may be obtained by fining the bilge diagonal, especially in the neighborhood where the parallel middle body joins the ends. In such cases, however, although there would be a virtual middle body, the actual middle body would more or less disappear. Further experiments upon this point are now under investigation. It may be of interest to note that in all the forms tested, where the ends were fine and accompanied by a somewhat abrupt change to the parallel middle body, the resistance at very low speeds always appeared to be higher than in those where the ends were fuller and the forms more gradual.

In any form there seems to be a certain combination of fineness of waterline forward and curve of sectional areas which will give the best result. In other words, in the neighborhood of the waterline it is of advantage to keep the form fine, but at some distance from the bow, there is what might be called a "governing section" to the form, which section should not be too full on the diagonal. Reference to the stream lines around models which were shown by Mr. Taylor last year, seems also to bear out the foregoing.

As a further illustration of this point, in the model of the F6 series with the full ends and no middle body (2B.2S), the same curve of sectional areas was held, but the fore body changed by fining the waterline and club footing the sections forward. The result at speed-length ratios of .75 to .85 was a diminution of the residuary resistance of about 9 percent; while in the form with the still finer waterline forward (1B.2S) the average over the same speeds was about 25 percent.

So far as the after-body is concerned, the most advantageous form appears to be one where the curve of sectional areas is of an easy character somewhat full on the waterline, and with an easy bilge diagonal.

The influence of shape of section upon the resistance when the curve of sectional areas remains the same, has not, however, been investigated, but some experiments are being made upon this point and will be submitted at a later date.

Speed in Warship Construction.

Few people realize when considering the noteworthy record which has just been made in launching the battleship *North Dakota*, 60 percent complete, in a period of ten and three-quarter months, that one of the oldest vessels in the United States navy was on the stocks nearl fifty years from the time of laying the keel to the time of launching. In 1817 the keel was laid at the Portsmouth (N. H.) navy yard for an 84-gun ship of the line, named the *Alabama*. This ship was not launched until 1864, when she was hurriedly completed for service in the Civil War. Also of six double-turreted monitors authorized in 1886-1887, with one exception, from seven to twelve years elapsed between the laying of the keel and the launching of the vessel.

A NEW GAS-ENGINE CYCLE.

BY ROBERT MILLER.

While investigating the subject of internal-combustion engines, the writer was impressed by the lack of agreement between the axioms laid down by Beau de Rochas in 1862 and his cycle brought to a working basis by Dr. Otto. Briefly, these axioms are:

1. Highest possible pressure at beginning of stroke.
2. The maximum speed of expansion.
3. Expansion to the lowest possible pressure.
4. The maximum cylinder volume with minimum wall space.

The present two and four-cycle engines fall short of these ideals for the following reasons:

1. The highest possible pressure for a given compression is not reached, because of imperfect scavenging.
2. The speed of expansion is limited to the piston speed, which must be kept down for mechanical reasons. In this respect the "free-flying piston" engine is ideal, but out of the question for modern service.
3. Expansion is not carried very far; pressures as high as 60 pounds per square inch are often rejected to the exhaust.
4. Maximum volume with minimum surface is counteracted by excessive wall cooling, made necessary by pre-ignition conditions and charge reduction.

To approach nearer the ideal, the writer devised the following cycle:

1. The entire cylinder, with clearance, is scavenged by an excess of pure, cold air, which therefore gives a higher initial pressure.
2. During the expansion stroke the combustion cylinder is connected to a larger supplemental cylinder with piston, so that the speed of expansion shall be faster than that with the smaller cylinder alone.
3. The supplemental cylinder brings the expansion to a low terminal pressure.
4. Cooling is required only for the purpose of lubrication, the fuel component being added during the compression stroke

The design of an engine embodying these features allows also some other very desirable ones: self-starting, reversing, double acting, etc., as treated more fully below.

DESCRIPTION OF THE CYCLE.

Referring to the drawings, we shall go through a complete cycle in one cylinder, the drawing showing a single-crank double-acting engine. Assume that in the clearance space *a* (upper cylinder) an explosion is about to occur. At that momen the valve *b*, being open, establishes communication between the combustion chamber *a* and the supplemental expansion chamber *c* (whose area is twice that of *a*), the valve *d* being closed. The result is that the force of this explosion is expended in the cylinders *a* and *c*, so that the expansion will be very rapid, and as the piston reaches the end of its stroke the volume of the combined cylinders *a* and *c* will be three times greater than that of *a* alone. The expansion must then have been carried down to about 5 pounds to the square inch pressure, with the result that, due to the rapidity of the expansion, little heat has passed through the cylinder wall.

While the differential piston was moving in this direction, having on one side the explosive impulse, on the other side it was compressing (to a slight degree) the pure air contained between it and the head; in the lower cylinder, of course, the crank case is used for this preliminary compression. As the differential piston nears the end of its stroke the valve *d* will open so that the remaining pressure in the cylinders *a* and *c* will escape through the exhaust port in this valve. As this pressure dies away, the slightly compressed air on the other side of the piston comes up through the connecting pipe, through the body of the valve *d*, and out through the ports *g*. As the pure air emerges from ports *g* it traverses the body of the cylinder and clearance *a*, goes through the valve *b*, the

chamber *c*, and discharges through the exhaust port of valve *d*.

The piston now starts on its return stroke. Valve *b* closes and the pure cold air in *a* will be compressed for the next impulse, the fuel being added during this stroke. While this new charge is being compressed, the supplemental chamber *c* is expelling its contents through the exhaust port in valve *d*, and the other side of the piston is drawing in a new charge of air, as in the ordinary two-cycle engines.

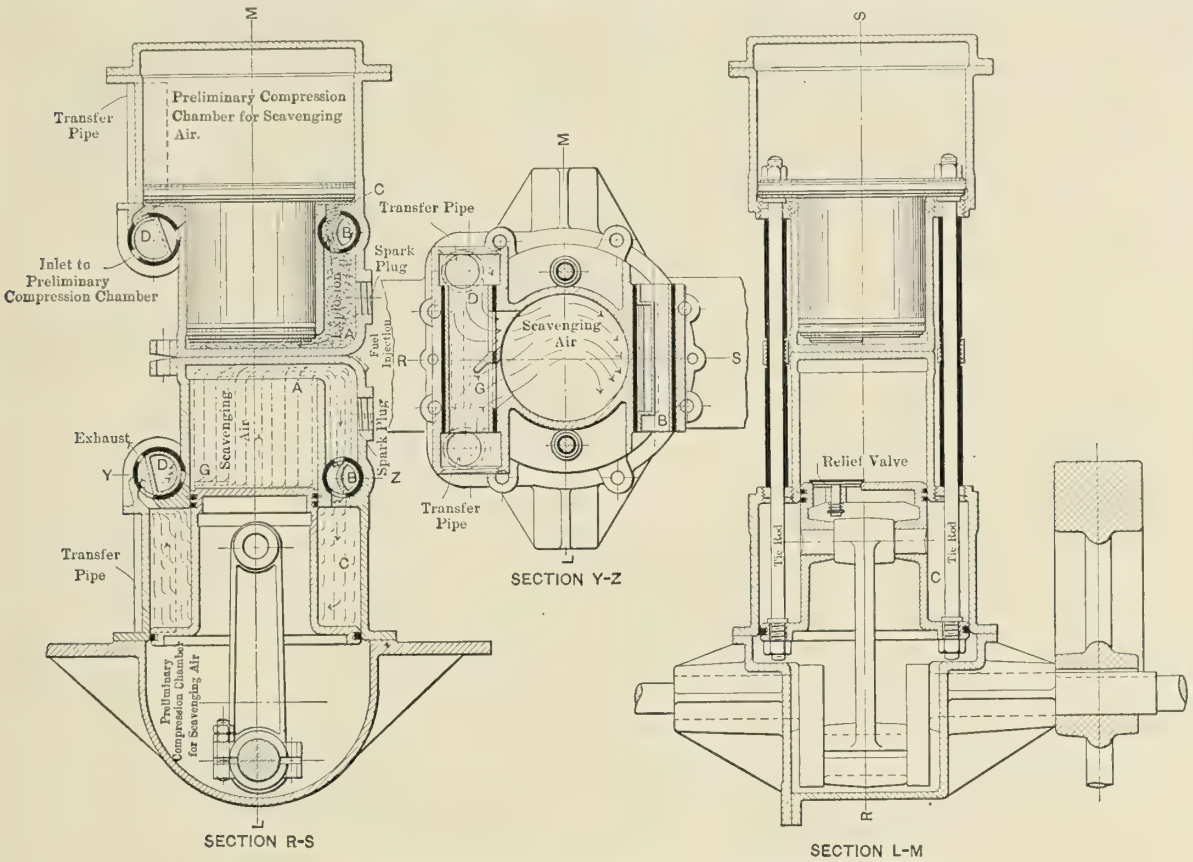
Just before ignition occurs valve *d* is closed and valve *b* gradually opened. As ignition occurs the cycle repeats.

To illustrate the increase in expansion speed, assume a piston speed of 900 feet per minute and a stroke of 12 inches. The ordinary cylinder will expand its charge from, say, 300 pounds to 45 pounds per square inch in one stroke, or in 1/900 minute. The Miller cycle expands the same charge, between the same limits, in one-third of 1/900, or 1/2700 minute. This method of increased expansion differs widely from the two commonly attempted. The true compound gas engine (two-stage expansion) fails because of the low specific heat of the working fluid, the transference of highly-heated gases through small ports and valves, which must be kept cool to insure durability, and the enormous amount of cooling surface in the high and low-pressure cylinders.

The other method of using a small charge (cut short on the suction stroke or expelling some on the compression stroke) involves a long stroke, and what is gained by increased expansion is lost by increased time of contact with water-cooled cylinder walls.

The gain with the Miller cycle in fuel economy and in lightness will be due to the conversion of more heat into work during the expansion stroke, instead of being wasted through the cylinder wall and the exhaust, the increased charge weight and mean effective pressure, because the new charge is pure, cold air at atmospheric pressure, and the addition of the fuel during the compression stroke.

The chief difficulty with the two-cycle engine is the time element, giving imperfect scavenging. In the large 1,000-horsepower European engines of this type it is usual to consider that scavenging commences at the outer dead center, because as the exhaust ports open the gases do not drop to atmospheric pressure instantly. The scavenging commencing at dead center, much valuable time has been lost, as this is the slow period of the stroke. With the Miller cycle, however, the results are vastly different. The pressure of the gases being brought down to a low point by the expansion cylinder, as the exhaust port opens, very little pressure drop is necessary before the scavenging charge enters. The result is that while the piston is approaching dead center, turning dead center, and returning to close the air port scavenging is going on. This large increase in the available time, so important with two-cycle engines, is of the greatest advantage.



SECTIONAL VIEWS OF THE MILLER SINGLE-CRANK, DOUBLE-ACTING GAS ENGINE.

Air cooling for marine service appears quite radical. But it can be shown that for medium-sized engines the internal cooling, as used here, is sufficient; whereas, for larger sizes the exterior portions could be cooled by the use of air jackets, through which a large volume of cold air could be drawn by an ejector operated by the large volume of the exhaust.

The internal cooling is, of course, due to the combined action of the method of scavenging and the lower wall temperature resulting from the mode of expansion. Other cycles hold the high temperatures so much longer that the heat flow into the walls must be much greater than with this cycle. Dugald Clerk has shown, experimentally, with a four-cycle engine that during the exhaust as well as during the compression stroke heat flows to the jackets. With the Miller cycle it is safe to assume that with the lower wall temperaures, resulting from the simultaneous expansion feature, the cold scavenging charge will prevent the head of the piston from accumulating too much heat, will chill the interior surface of the cylinder bore, where the oil is, cool the valves, and keep

within limits the temperature of the surface of the expansion cylinder. Cooling is really required in this engine only for the purpose of lubrication.

It must not be forgotten that the injection of the fuel in the liquid form also has its effect in lowering the temperature of the compressed air. Being a liquid it must, to assume the gaseous state, abstract heat from its surrounding medium.

Not the least of the advantages with this method of cooling is the elimination of much pattern and foundry labor, the freedom from danger of cracked cylinders in cold weather, and the doing away with troubles due to the deposits of saline matter in the jackets, with consequent overheating.

While the method of adding the fuel component to the charge by injecting it during the compressing stroke is not new, yet as applied to this cycle it is in results somewhat different from that usual in most American oil engines. The common method of fuel injection has the disadvantage that much stratification occurs, so that the mean effective power, which should be higher, because of the increased charge weight, does not come up to expectations. The Diesel, Haselwander and other Continental engines break up the fuel stream by compressed air, thus giving an intimate mixture.

With the Miller cycle the two-charge components are thoroughly mixed by the blast of air resulting from the opening of the valve *b*, establishing communication between the combustion and expansion chambers. It is to be remembered that in order to prevent a free expansion drop when the two chambers are put into communication just before dead center, the exhaust valve *d* of the expansion chamber closes, so that the remaining gases trapped therein (largely pure air) are compressed, because of the very small clearance, to a degree somewhat higher than that existing in the combustion chamber. This has the effect also of balancing the valve *b*. In fact, this phase of the cycle might be worked out to act as a timing valve with automatic ignition.

Regarding the desirability of fuel injection over the usual carburetor methods a few words might be said. For naval and commercial vessels the item of reliability will be paramount. The vagaries in fuel compounding with carburetors induced by thermometer, barometer and hygrometer changes, and the problems involved in requiring a gaseous medium, air, to conform to the laws of liquid fuels may be tolerated in pleasure service, but hardly elsewhere. It is significant that the great majority of kerosene and heavy-fuel engines used all over the world, even with ignorant help, is of the injection type. While on the subject of reliability, attention is called to the fact that with the elimination of the exhaust gases from the new charge and with fuel injection, increased flexibility and certainty of operation are bound to follow.

Semi-rotary valves are shown; poppet valves could be used just as well, but the former possess so many advantages in quietness, ease of operation and long life that it is thought, in view of the lower wall temperatures possible with this cycle, that the difficulties arising with rotary valves in the ordinary cycle will be absent here. In fact, the largest modern air compressors make use of this type of valve.

The prevailing practice with large marine and railroad engines for starting seems to be by compressed air stored in a tank and supplied by a small attached compressor. The writer has always believed that this method with its complications may be unnecessary and that the gas engine possesses in itself the means for self starting.

A mixture of gas or gasoline vapor and air ignited at atmospheric pressure results in explosion pressures, ranging from 30 to 95 pounds per square inch. If a supplementary chamber be connected to the cylinder and the entire contents of the combined volumes be ignited at atmospheric pressure, the explosion proceeding from the chamber to the cylinder compresses the unburned mixture before it (Clerk Lanchester Method) and pressures up to 250 pounds are made available.

With the corburetor system of fuel compounding, of course, when the engine stops the cylinders will be filled with mixture. But gasoline vapor not being a fixed gas, as the cylinders cool the vapor will condense out of the mixture, so that the possibilities of self starting depend upon this phenomenon. A gasoline engine should be capable of starting at any time, after stopping, so that this method is out of the question.

The Miller cycle has all the advantages and more of the Clerk Lanchester method. Instead of a supplementary chamber the expansion chamber *c* is utilized and the resulting explosion acts not only on the contracted portion of the piston, but on the the larger portion as well. The two chambers being in communication and the spark being made, it is only necessary that the gasoline be squirted in as a fine stream toward this spark. The single-crank double-acting type stops by cutting off the fuel supply, and combustion and expansion cylinders must therefore be filled with pure air, which still will be pure air a week or a month later.

The engine would hardly stop on dead center, because that would be against the full compression; in either cylinder the crank position would therefore be somewhere about mid-stroke. The resulting impulse would be amply strong to turn the engine several revolutions. Fuel injection being a positive factor, the engine must take up this cycle before one-half a revolution is passed.

Much that had been said of self starting applies to reversing. While the arrangement of the valves in the Miller engine is such as to permit easier application of compressed air, yet, as has been shown in the paragraph on self starting, the engine can be made to reverse just as positively without extraneous devices. Here, again, the expansion cylinder comes into play. The valves are operated by eccentric with a Stephenson link motion. Suppose that the engine is turning over in the forward direction, the explosion taking place in the top cylinder, and the rotation is to be reversed. The link is pushed over so that the reverse eccentric comes into action. The lower exhaust valve *"d"* is closed instead of being opened, and the upper valve *"d"* is opened instead of being closed. The upper cylinder loses its impulse, the lower cylinder traps the exhaust gases in the expansion cylinder; the motion of the engine is slowing down on an air cushion, as these gases are gradually compressed until a point is reached where the compression overcomes the forward motion.

The engine reverses by reason of this compression, aided by the explosion tripped prematurely.

Should the engine be required to start in the reverse direction after standing idle for sometime, the self-starting operation, as sketched above, is gone through with the links in the reverse position.

The pistons traveling together are connected by tie rods. To prevent leakage, long sleeves are used. These sleeves, because of their considerable length and oil packing, are very efficient and are exempt from the usual stuffing-box troubles. The tie rods themselves are lapped in the sleeves and are always in tension. They operate in the comparatively cool atmosphere of the expansion chamber. Where necessary, the sleeves could be subjected to a blast of cold air from the ejector action of the exhaust, as explained before.

As applied to naval and high-speed vessels, the absence of heavy auxiliaries, the high mean effective pressure, the frequency of impulses should result in light weight. It is not unreasonable to suppose that in large units an average weight of one pound per horse-power could be obtained.

A phase not unimportant is that with the use of pump injection the possibility of back firing in the carburetor is eliminated, with consequent lower insurance rate. Alcohol, kerosene and other safe oils operate best with pump feed, and once started with a small quantity of gasoline or alcohol alone should give no trouble.

SIXTEENTH ANNUAL MEETING OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

The sixteenth annual meeting of the Society of Naval Architects and Marine Engineers was held in the Engineering Societies Building, New York, November 19 and 20, 1908. The report of the secretary-treasurer shows that the total receipts for the year amounted to $9,756.19 (£2,003) and the disbursements, $8,305.49 (£1,705). The total resources of the society at the end of the fiscal year were $27,818.51 (£5,712), with no liabilities against it. The membership at the beginning of the year was 828. This has decreased during the year to 765, but, by the election of forty-six new members at the present meeting, the total membership is brought up to 811.

PAPERS READ NOVEMBER 19.

No. 1—The War Eagle.—1904.

BY CHARLES H. CRAMP.

Chesapeake Bay in about the first half of the last century was noted as the home of two classes of vessels known over the entire world as the "Baltimore clipper" and the "pungy." Conditions of commerce at that time were such that high-speed sailing vessels were required, and both of these types were world famous in this respect. On the Chesapeake the "pungy" was always built "by the eye." There was no model, no "laying down" or "laying off" in the mold loft, nor were molds made there from a "body plan." Each builder would have a stem and a midship-section mold for all vessels, and as there was little difference in the dimensions of the boats, these two molds could be adjusted to an increase of a few inches in depth or width to suit contract requirements.

Having built a celebrated "centerboard" schooner named the *Wood Duck,* for a party in the oyster trade in Philadelphia, another party, a rival in the same business, placed a contract with my father to build a "pungy" that would be the fastest in both bays (the Chesapeake and Delaware). This vessel, named the *War Eagle,* was the only one of her type built north of Baltimore with the exception of a pilot boat or two, and easily fulfilled the stipulation in the contract that she should be the fastest in the two bays. The sailing qualities of the boat were remarkable and she was never beaten either in a match race or when plying her usual trade.

There was but little difference in the model between the "pungy" and the pilot boat of our Atlantic ports and in general design of the "cup-defending" yachts. The fast ship, whether it was in the form of "pungy," "clipper," the "rover," or the "long, low, black schooner," was particularly an American product, and the methods in its production were born in the American shipbuilder at that time. No model in any country for form or seaworthiness has ever equaled that of fifty years ago.

The Oldest Iron Ship in the World.*

Sixty-six years ago there was dragged across the mountains from Pittsburg to Lake Erie the plates and angles, bars and beams, bolts and rivets, forgings and castings, out of which was constructed what is now the oldest iron ship in the world. This is the United States man-of-war *Michigan* (rechristened in 1905 the *Wolverine*), which since that time has continued to be the whole United States Navy on the Great Lakes, where treaty obligations restrict naval representation to one ship on each side. She was a veteran when Ericsson's *Monitor* threw all the navies of the world into the junk-pile, and was forty years old when the first ships of our so-called new navy were authorized, and they in turn have gone into the discard long years since.

She is rated as an unarmored, unprotected cruiser, with a water-line length of 165 feet; beam, 27 feet; displacement, 685 tons; indicated horsepower, 365; speed, 10.5 knots (12 statute miles); coal endurance at 10 knots, 2,240 nautical miles, or 224 hours; bunker capacity, 100 tons.

Much of our steam engineering knowledge is based upon the early experiments on expansion of steam, superheating, steam-jacketing, condensation, etc., carried out on her by B. F. Isherwood, afterwards Engineer-in-Chief, United States Navy, when attached to her. The value of those experiments, and the theories deduced from them, to the world at large, are beyond computation, and they are to this day quoted as authority. The engines are of the simple inclined type, with two cylinders 36 inches diameter, 8 feet stroke, and although of an antiquated type, their condition is stated in the Bureau reports to be good. The original paddle-wheels were 21 feet 10 inches diameter and had each sixteen paddles or buckets. The dip of the wheels was 32 inches and the maximum revolutions 22 per minute. The draft was 7 feet 8 inches. The boilers were of the "box" type, 8 feet 6 inches wide, 19 feet long, and 9 feet high, and the working steam pressure was 15 pounds per square inch. These old boilers have long since been replaced by others of a modern type, and new paddle-wheels of smaller diameter are also fitted. The draft is also increased to 9 feet, no doubt partly due to the increased boiler weights and partly to increased coal capacity, made possible by shorter boilers. But the hull and engines and most of the equipment, except armament, remain as they were built sixty-six years ago.

The *Wolverine* has not only a sentimental value but she is in active service, and she has contributed much to material progress, and as she is likely to be useful for many years to come, the most graceful thing the Department can do, the greatest compliment it can pay, not only to the old ship herself but to our neighbors across the line, is to continue her in commission.

No. 2.—Practical Methods of Conducting Trials of Vessels.

BY COL. E. A. STEVENS.

ABSTRACT.

Speed trials outside the navy are in this country only too often carried out under conditions that render the results "painfully inaccurate." These conditions are due to many causes, but chiefly to a lack of appreciation of the value of the data to be obtained, to an exaggerated idea of the cost, and sometimes to carelessness.

Where only a limited number of runs are available the trials should be planned with care and run as systematically as possible. The apparatus needed is neither very expensive nor intricate. It should, however, be reliable. The observations, after all, can give knowledge on but three points:

1. The time on the course.
2. The revolutions—either the total on the course or the revolutions per minute.
3. The mean effective pressure in cylinders.

The record of time and revolutions should be taken simultaneously. The device illustrated in the paper by the late Naval Constructor Woodward (see Transactions of 1905), furnished an excellent means to insure simultaneous records and a very accurate registering mechanism. A device I have used for this purpose with much success consists of a contact arranged to work from the indicator-gear lever, which opens and closes a circuit at each revolution. A magnet is energized by the current, and moves a plunger against a spring which, on the opening of the circuit, forces the plunger back. This motion is recorded on a counter. The counter is started by a switch at the beginning of each run and stopped at the end; a watch held by the same observer can be started and stopped

* Discussion of paper No. 1, contributed by Henry Penton.

at the same time. The result is, of course, not as accurate as the device suggested by Woodward but is much less expensive.

Stop-watches used on trials should be carefully rated. Indicators are often incorrectly rigged. Their springs should be tested. A tachometer set in plain view of the man at the throttle will enable him to maintain a constant speed, and will also furnish a check on the revolutions per minute at the time that indicator cards are taken.

A very important point is the determination of the speeds at which runs shall be made. The settlement of this depends on the system on which the results will be plotted. To properly determine a curve of speed on revolutions per minute, five spots should be had at different speeds. These spots can be obtained by the second means of three runs at each speed, or more closely by the mean of second means of four runs at each speed. This involves twenty runs, which is usually more than can be had.

If the system in use in the navy, which is based on a set of runs at evenly progressive speeds with four runs at the highest speeds, is employed, the runs must be made at evenly progressive speeds and at nearly equal time-intervals. Failure to comply with these conditions may induce considerable error.

To meet ordinary conditions the tidal formula explained in the Transactions of 1901 seems generally preferable. This formula is based on the equality of advance per revolution. Fairly accurate results can be had with ten runs and good results with twelve. Runs can be made in any order of speed provided enough pairs be made to derive tide curve. Although the use of this formula is more laborious than plotting the runs separately, the amount of labor is not excessive.

Whatever system is adopted it is well to remember that our observations are accurate only within a certain limit. That limit may, by careful observation and expensive apparatus, be reduced, but it cannot be eliminated. The error involved where ordinary apparatus is carefully and intelligently used is not material from the designer's standpoint.

No. 3.—The Influence of Midship-Section Shape Upon the Resistance of Ships.

BY D. W. TAYLOR, NAVAL CONSTRUCTOR, U. S. N.

(This paper is published in full on page 525.)

No. 4.—Further Experiments upon Longitudinal Distribution of Displacement and Its Effect upon Resistance.

BY PROFESSOR HERBERT C. SADLER.

(This paper is published in full on page 530.)

No. 5.—Further Propeller Analysis.

BY CLINTON H. CRANE.

Assuming that the law of comparison holds good for propellers, let T = thrust of any propeller; p = pitch ratio; D = diameter; R = revolutions per minute; s = slip ratio; H = the brake-horsepower required to turn propeller of diameter D with pitch ratio p and slip s; and T_1, D_1, R_1, H_1, represent the thrust, diameter, etc., for a similar propeller working at the same slip, and let S = the power factor in "Taylor's" formula.

Then $\frac{T}{T_1} = \frac{D^4 R^2}{D_1^4 R_1^2}$; and $\frac{H}{H_1} = \frac{D^5 R^3}{D_1^5 R_1^5}$.

Let D_1 = unity and R_1 = 100.

Then $T = \frac{D^4 R^2}{10{,}000} T_1$; and $H = \frac{D^5 R^3}{1{,}000{,}000} H_1$.

The variable T_1 has been obtained by Prof. Durand for a series of propellers, and also a variable which we will call w_1, or work in foot-pounds per revolution at 100 revolutions per minute for a propeller 1 foot in diameter.

H_1 will $= \frac{100\, w_1}{33{,}000}$, and H will $= .00000000\ 303\ D^5 R^3 w_1$

Naval Constructor Taylor gives $H = .0093648\ D^2 V^3\ S$.

Now $V^3 = \left(\frac{p D R (1-s)}{101\ 1/3} \right)^3$;

$H = .000000009\ D^5 R^3 p^3 (1-s)^3 S$; and

$w_1 = 2.97\ p^3 (1-s)^3 S$.

This gives an easy method of converting w_1 into S or S into w_1. As w_1 gives us a variable which is practically of the first degree, and S a variable of the third degree in terms of slip, it is preferable to use w_1. w_1 increases with the pitch ratio, with the thickness of the blade, area of the blade, and with the slip with all propellers except those of very small pitch ratio.

From a careful study of values of w for propellers of the same blade contour and blade thickness and the same slip within reasonable changes of pitch ratio

$$\frac{w}{w_1} \text{ will} = \left(\frac{p}{p_1} \right)^{3/2}$$

It will be noticed that the horsepower formula deals with the fifth power of the diameter, the third power of the revolutions and the first power of w, consequently in determining diameter from this formula a small error in w produces a very much smaller error in diameter; that is, an error of 10 percent in w would only give us an error of 2 percent in diameter.

No. 6.—Deviation of the Compass Aboard Steamships—Its Avoidance and Correction.

BY L. H. CHANDLER, LIEUTENANT-COMMANDER, U. S. N.

This paper is of such great length and treats the subject in such a complete manner, beginning from fundamental principles and outlining in detail the many different ways of compensating for the deviation of compass on board steel ships, that space will not permit an abstract which would adequately cover the subject-matter which it contains. The paper is a statement of the general mathematical principles involved, illustrated from the results of observations taken and worked out by the navigators of the vessels of our battleship fleet. As the members of the society have never given much attention to the question of the magnetic effect of the structure of a steel ship upon the compass, the author took occasion to present this paper in the hope of arousing an interest which will lead ship designers and builders to realize the necessity for a clear comprehension of the general principles involved, and the need for careful planning in arranging the details of construction of vessels in the vicinity of compass positions. A little lack of understanding or care in regard to this point on the part of the designer and builder often places the navigator, from the start, under a handicap so severe in its nature as to often constitute a real and unnecessary menace to the safety of the ship.

After briefly describing the mariner's compass and the effect of the earth as a magnet on the compass, variation of the compass is defined, and the forces acting upon a ship's compass, including the total magnetic force of the earth, and the magnetism of the adjacent members of the ship are described. Following this is a comprehensive mathematical discussion of the relation of the magnetic forces affecting a ship's compass, including the derivation of Poisson's equations. Having discussed, from a theoretical point of view, the forces which cause deviation, and having deduced certain equations concerning it, the principal practical methods employed by navigators in determining, correcting or compensating for it are considered.

First, methods of compensation of a new ship alongside a dock are available. Having obtained an approximate compensation by one of these methods, an exact compensation can be made while the ship is under way. Throughout the first method it was assumed that the sub-permanent magnetism was really permanent, but, when the ship is actually steaming at sea, this may change appreciably from day to day, especially in a new ship, as the magnetism absorbed in building "shakes out." For this reason the presence of iron near the compass is bad, not only because of the initial error it causes, but because a very small change in its magnetic condition produces a greatly exaggerated change in the compass, and so introduces a most uncertain element of danger. It is just here that the designer and builder, who appreciates the importance of these effects, can best help the navigator by changing the position of certain parts of the structure, so that they will have a less effect upon the compass.

Finally the subjects of perfect theoretical compensation for semi-circular and quadrantal deviation are discussed and it is pointed out wherein they fail in actual practice. A vast amount of valuable data on the results of the compass observations taken on board the battleships and torpedo-boat destroyers of the Atlantic fleet during its trip around South America last winter are given. There are some gaps in the data, and some results that are probably in error, as the observations were taken and the data worked up by officers whose time was fully occupied in the performance of their daily duties. Even with these omissions and errors the results are of inestimable value.

No. 7.—The Influence of Free Water Ballast Upon Ships and Floating Dry-Docks.

BY T. G. ROBERTS, NAVAL CONSTRUCTOR, U. S. N.

ABSTRACT.

This paper contains a mathematical consideration of the theories governing the various considerations of stability of a floating dock throughout its various operations. In considering the loss of stability, due to free water ballast, it is shown that the moment of stability of the ship is equal to the righting moment of the ship minus the capsizing moment, due to the free-water surface. On the other hand, it is evident that there may be an increase of stability, due to free water ballast, since the free water may swing farther to either side of the vertical than the roll of the vessel, and its action may either aid or hinder the rolling between the extreme values of the shifting moment applied positively or negatively. In a floating dock the longitudinal stability must be provided for by a sufficient number of longitudinal compartments. But usually structural considerations require a greater number of such subdivisions than necessary to compensate the longitudinal stability. The transverse stability is much more sensitive and will need transverse subdivisions for compensating the loss of stability, due to free water surface. The mathematical considerations for finding the most suitable number of longitudinal compartments of equal width, so that the metacentric height will not be materially reduced, gives some definite idea of the value of additional compartments, showing that, after a certain necessary number additional ones may be of little value.

To find the interior water depth for a given immersion, it is noted that the displacement of the dock at any draft is equal to the weight of the dock plus the weight of the water inside the dock. It is evident, therefore, that the outside waterline is higher than the inside water surface. The method of locating the corresponding positions of the two consists of plotting curves of displacement of the dock at various drafts, and curves representing the weight of the interior water corresponding to the various water-level heights. From these a curve showing the total weight of the dock and interior water for various water levels is obtained. If a ship is in the dock, the curve of the ship's displacement must be added to the curve of the dock's displacement and weight of the interior water.

A consideration of the stability of the dock requires calculations when the dock is free under all different conditions, and also when a ship is in the dock under all conditions. The mathematical discussion leads up to a formula for the coefficient of stability in which all values occurring having been obtained by direct calculation, and expressed in curves, corresponding values may be taken from the curves at any draft. The greater the weight of the ship in the dock, the less will the coefficient of stability become. As the dock inclines, the stability increases with the angle of heel with the dock free, and by a greater proportion with a ship in the dock. Therefore, the stability is a minimum at the vertical position for any probable angle of heel, and the stability curve increases as the dock inclines by righting moments greater than for the displacement cut by the ship in dock inclined at the draft of the instant under consideration.

The forces tending to bend or break a floating dock act in the same manner as for a ship considered as a floating beam. It is customary on the part of some to consider that the side walls of a dock should be built with sufficient girder strength to carry the entire load. As a matter of fact, the load rests upon the center pontoons, and is transmitted through the connections to the side walls. All portions of side walls, pontoons and connections carry each its own share of the distributed load, both longitudinal and transverse. The strength calculations, therefore, consider the entire dock as a single beam, longitudinally or transversely. From the curve of bending moments it is found that, with a ship docked and the dock pumped dry, if we introduce water into the end pontoons the maximum bending moment at the midship section will be reduced. In a certain dock it was found possible to so flood the ends with a depth of 8 feet 9 inches of water inside, and still leave enough freeboard at the pontoon deck platform. In that condition the unit stress at the most strained section was reduced to less than one-third its former value. It is therefore wise to retain as much water as possible in the end pontoons in docking a large ship where the stresses are great, especially when the ship is to remain in dock for a length of time.

No. 8.—Some Recent Inventions as Applied to Modern Steamships.

BY W. CARLILE WALLACE.

ABSTRACT.

For the safety of the crew and for convenience in working the vessel, an arrangement possessing the following qualifications by which every bulkhead door in the vessel can be closed from the bridge in a matter of a comparatively few seconds should be installed on every ship. First, it must be possible under ordinary conditions at sea or in port to open and close the doors at will, leaving them either open or shut. Second, before the doors are closed from the bridge an automatic warning must be given to those below, after which they must close slowly, not drop suddenly. Third, after the doors are closed from the bridge it must be possible to open any individual door, the door closing automatically again. Fourth, in the event of water entering any compartment to a dangerous extent, the doors in immediate proximity to this compartment must close automatically. And last, the means adopted for closing the doors must be such that even when the mechanism is submerged it will still perform its work.

Four mediums are available for doing this work: steam, air, water and electricity. Steam is inadmissible chiefly on account of bursting steam pipes. The pneumatic system has

been tried and found wanting, to say nothing of its being too expensive. Electricity has been used with considerable success in this country, but has objections, which appear to the writer to relegate it to the second place. Among them is the possibility of blowing out of fuses or injury to the motors through overload, the risk of short circuit should the gear or conductors become submerged, and the great difficulty of locating or remedying a fault in the system. For these reasons hydraulic control seems to fulfill, more nearly than any other, the requirements necessary to a thoroughly reliable water-tight door-closing system. In the writer's opinion the hydraulic system in use on the two large Cunarders fulfills every requirement of a perfect water-tight door system.

The other inventions considered tend more toward the comfort of passengers than to their safety. The first of these is the new means of disposing of the ashes and clinkers from the stokeholds of vessels without the necessity of hoisting them above the main deck and dumping them over the side; or forcing them above the waterline by a jet of water and then over the side through a bent pipe. The new method consists of an apparatus by which the ashes and clinkers are forced through the bottom of the ship by means of compressed air. The expeller proper consists of a hopper to receive the ashes and clinkers opening into a crusher, which breaks up the large clinkers. Below the crusher is a drum revolving horizontally in a water-tight casing. As it revolves the inside of the drum is alternately in communication with the chamber below the crusher, and the discharge opening through the bottom of the ship. While in communication with the latter, compressed air is admitted, expelling the ashes through the bottom of the ship.

Two other recent inventions of importance include a device for cooling state rooms in vessels trading in the tropics and fitted with refrigerating machinery, and a device for maintaining electrically-heated staterooms at a definite temperature irrespective of atmospheric conditions. The cooling device consists of a pipe containing a brine coil supplied from the refrigerator, through which the air supplied to the stateroom is drawn by a small motor-driven centrifugal fan. The other device is the Geissinger electric thermostat, by means of which electric heaters are automatically controlled to maintain a room temperature constant within the limits of a few degrees, thus saving electric energy and adding to comfort.

PAPERS READ NOVEMBER 20.

No. 9.—Service Test of the Steamship Harvard.

BY PROFESSOR C. H. PEABODY, W. S. LELAND AND H. A. EVERETT.

(This paper is published in full on page 522.)

No. 10.—Trials of the U. S. Scout Cruiser Chester.

BY CHARLES P. WETHERBEE.

ABSTRACT.

The final acceptance trials of the United States scout cruiser *Chester** took place on Oct. 20 and 21, 1908, in a heavy northwest gale, with the following results:

Duration, hours	12	4
Displacement, tons	3,800.	3,630.
Mean revolutions per minute	479.6	590.1
Corresponding speed, knots	23	26.1
Miles per ton of coal	2.3	1.5
Average air pressure, inches	0.75	2.75
Coal per E. H. P. per hour, pounds	3.61	3.23
Coal per I. H. P. per hour on basis of 0.55 propulsive coefficient	1.97	1.78

* For a general description of this vessel, see page 254 of our June (1905) issue, and for a record of the preliminary acceptance and standardization trials covered in the first part of this paper, see our issue for September, 1908.

On these trials in a very rough sea the turbines did not race. The ship was operated on these trials by her regular navy crew under service conditions. The coal was ordinary run of mine taken from the Government pile at the Boston navy yard. The air pressures carried were, by direction of the board, regulated to be the same as on the corresponding preliminary acceptance trials.

The conclusion drawn by the writer from the trials of this ship is, that the combination of bent-tube boilers with rugged scantlings, or of other boilers capable of a high rate of forcing and evaporation per unit of heating surface combined with steam turbines, permits a lightness of machinery installation without any sacrifice of durability and reliability never before contemplated in our warship designs. Light machinery installations have been constructed in the past for torpedo boats and destroyers, but through the use of short-life boilers of very light scantlings and delicate parts in the main engines and auxiliaries durability and reliability have been sacrificed. The machinery of the *Chester* is not of this class. Lightness per unit of power is obtained through thermodynamic efficiency and high output per unit of surface. The boilers are of rugged design and have large, thick tubes. Their casings are heavy and durable. After 14,000 mile of steaming, one small part of the casing only has shown any effect of service. A slight thickening at this spot, and the addition of four small baffles about 6 inches square in the soot-boxes at the bottom of each boiler, will remedy this difficulty. Furthermore, the boilers can readily be cleaned. These boilers can be forced to evaporate 11 to 11.5 pounds of water per square foot of heating surface under gage conditions (not from and at 212 degrees). They are not light per unit of surface, but their ability to indefinitely withstand heavy forcing and show good efficiency makes them extremely light per unit of power developed.

Machinery of the *Chester* type is perfectly suitable for our battleships and armored cruisers. Its adoption will give our country ships that are a knot faster than foreign ships of the same displacement without any increase in machinery weight. Turbines have already been adopted in our latest designs, but while boilers adopted in our latest designs are strong and durable, they are not capable of the high output per unit of surface that can be obtained from the type on the *Chester*. It seems as if this advantage were worth serious consideration.

No. 12.—Shipbuilding on the Great Lakes.

BY ROBERT CURR.

This paper gives a detailed statement of the method of procedure in building a modern lake vessel of the hopper type from the ordering of the material through the laying out, fitting up and finishing of the ship.

No. 13.—The Steamer Commonwealth.

BY WARREN T. BERRY AND J. HOWLAND GARDNER.

ABSTRACT.

The *Commonwealth*, built for night service between New York and Fall River, is 455 feet 2 inches long over all, 55 feet beam molded, 22 feet molded depth with a draft of 13 feet at a displacement of 5,410 gross tons and 15 feet at a displacement of 6,410 gross tons. The maximum designed speed is 22 statute miles per hour. Experience with propeller and side-wheel steamers, in this particular service, demonstrates that very much greater overhang of guard can be fitted to the side-wheel boat. On the *Commonwealth* 19 feet 8 inches overhang, each side of hull, was necessary to provide for the spacious saloons and staterooms demanded by the traveling public. This extreme breadth of guard is in addition a very efficient safeguard against serious damage to the hull in case of collision. The compound inclined engine with feathering wheels, as adopted, combines the ability

to stop and back very quickly, utilizes only lower hold space, which is of very little value for freight or passenger accommodation, and avoids the excessive vibration common to screw propellers in shallow draft vessels.

The propelling machinery, designed for a maximum indicated horse-power of 10,000, is composed of a double compound, inclined, reciprocating engine with two high-pressure cylinders 50 inches diameter and two low-pressure cylinders 96 inches diameter, with a common piston stroke of 114 inches connected to two pairs of cranks set at right angles. All cylinders are fitted with double poppet valves, Sickles adjustable cut-off on the high-pressure cylinders, and Stevens fixed cut-off on the low-pressure cylinders, all operated with Stephenson links controlled by a 20-inch by 24-inch steam reversing engine. Two surface condensers of cylindrical type, each containing 8,000 square feet cooling surface, are located outboard of the low-pressure cylinders, with suction pipes to two vertical air pumps 5 feet diameter by 30-inch stroke connected to the low-pressure crossheads. The wheels are of the feathering type, 33 feet outside diameter. Steam is supplied by ten Scotch boilers located five on each side, each 15 feet 6 inches in diameter by 13 feet 6 inches long, having three Morrison furnaces, 50 inches inside diameter, with a total grate surface of 937 square feet and a total heating surface of 29,340 square feet. Forced draft is supplied to closed ash pits by four blowers. The electric outfit is comprised of two 75 K. W. generators with 10½-inch by 18-inch by 8-inch engines and one 50 K. W. generator with engine 9½ inches by 15 inches by 6 inches, located in the engine room on the main deck, supplying about three thousand incandescent lights, a 24-inch searchlight on top of pilot house, an electric elevator with a capacity of 2,000 pounds, and two electric blowers for the ventilation of the aft cabins.

Particular attention has been paid to fire protection and fire-fighting appliances. All wood work throughout the cargo space, emigrant quarters, crew's quarters on the main deck, and kitchen on the dome deck, is covered with galvanized iron, fastened directly to the wood. Steel decks are fitted over the machinery space. The engine-room and boiler-room ventilators and enclosures are of steel. Two fire bulkheads are provided, dividing the vessel into three fire compartments. Suitable sliding doors are provided in the main corridors and freight space. An iron bulkhead extending entirely across the upper deck house is fitted just forward of the kitchen range. There are sixty fire hydrants located throughout the steamer, with a complete equipment of portable hose and fire extinguishers. In addition to this an independent sprinkler system is provided, connected to manifolds in the engine room on which a pressure of 100 pounds per square inch is always maintained. Supplementary to this system is a thermostat system with mercury thermostats located not over 12 feet centers with all wires run in conduit, and divided into circuits corresponding with the sprinkler circuits.

In size, magnificence and completeness of equipment the *Commonwealth* represents the highest development of this type of vessel. The contractors were the Quintard Iron Works Company, of New York, and the hull was built at the yards of the William Cramp & Sons' Ship and Engine Building Company, of Philadelphia.

No. 14.—Centrifugal Pump Fire-Boats.

BY CHARLES C. WEST.

ABSTRACT.

The fire-boat is now rapidly coming into use as an auxiliary water supply to the municipal pumping plants in cases of large conflagrations. The pressure maintained and the volume supplied by the ordinary city water supply are insufficient in extreme cases and most of the large cities are now installing the system of water mains drawing their supply directly from the water front. Suitable connections to auxiliary mains are provided where these mains lead to the river. The feasibility and practicability of this high-pressure main system have been demonstrated, and consequently, in future, higher pressures and more capacity will be required of the pumps on the fire-boat.

The development of the centrifugal pump of the multiple stage variety is comparatively recent, and the advent of the steam turbine has aided in its adoption owing to the high speed obtainable. In the boats equipped with turbines and two-stage centrifugal pumps, there has been no difficulty in obtaining higher pressures than are possible in the reciprocating pump. This pressure can be easily increased by increasing the number of stages, with no effect on the structure or working of the pump. The type of turbine used in operating these centrifugal pumps has been the horizontal impulse type. This type of turbine is more compact than the reaction turbine, and it is claimed that it can be started without being warmed up, a performance said to be dangerous in the reaction turbine owing to the liability of warping the spindle.

Two 9,000-gallon boats recently built for the city of Chicago have the following dimensions: Length over all, 120 feet; beam, 28 feet; depth, 15 feet; mean draft, 9 feet 6 inches, and a displacement of 500 tons. The main pumping and power machinery consists of two 660-horsepower Curtis turbines direct connected to 200 K. W. direct-current generators and two-stage centrifugal pumps. The generators served to provide current for the propelling motors, which are of the variable speed-reversing type. There are two pumping-generating sets in each boat, and twin screws. Steam is supplied by two two-furnace Scotch boilers, with total of 3,820 square feet of heating surface and 84 square feet of grate surface. Forced draft is provided on the closed stoke-hole system. The pumps, though usually run singly, can be compounded, and a nozzle pressure of over 300 pounds was thus obtained on a test, with a resultant capacity of about 5,000 gallons per minute.

In an 8-hour test on one of these boats, 4,800 gallons of water were discharged per nozzle per minute, or a total of 9,600 gallons for the boat. The steam consumption of the turbines per B. H. P. per hour figured out at 18.4 pounds and the coal consumption per B. H. P. per hour at 2.82 pounds.

The data obtained from the foregoing test show not only that the centrifugal pump is a more powerful and reliable machine than the old type, but also that the greater economy of the turbine outfit makes it possible to do more work with about half the boiler capacity. Comparing the water rate of these turbines with that of a high-pressure pump taking steam for almost full stroke demonstrates the fact that there is probably no boat afloat of the same size that has the same pumping capacity.

No. 15.—Sea-Going Suction Dredges.

BY THOMAS M. CORNBROOKS.

ABSTRACT.

Recently, the Engineer's Department, United States Army, have contracted for a number of sea-going suction dredges of varying sizes, ranging from 166 feet to 300 feet long, the capacities of bins ranging from 1,000 to 3,000 cubic yards. The dredged material is carried in two large bins, one forward and one aft of the machinery space. The officers and crew have commodious quarters in houses on deck and on the lower deck forward and aft. The propelling machinery consists of two compound engines, steam for which is furnished by four single-ended boilers. A very large rudder is necessary, due to the shafts being so far outboard and the extreme fullness of after body.

In operating, the dredge is kept moving forward, at a speed of about six knots, with the suction pipes dragging on the

bottom. The material is sucked up by 20-inch centrifugal pumps and discharged into bins, through pipes and distributing chutes on top of the bins. By means of gates in the bottom and sides of these chutes it is possible to distribute the material evenly. As the bin fills the water is drained off by overflows through the sides. When the bins are filled the dredge proceeds to the dumping grounds and, opening the gates in the bottom, drops the material. The gates are operated by means of a double-cylinder vertical engine through worms and fixed nuts on vertical rods. The percentage dredged depends on the quality of material and ranges from 10 to 60 percent.

The curves of bending moments, etc., and equivalent girder, for the latest New York harbor dredges show that the neutral axis of the girder is very low, showing that the material is not distributed to the best advantage. The tension is the sheer strake works out at 4.22 tons and compression in the keel at 2.73 tons.

On two dredges, built for the Isthmian Canal Commission, the deflection under load measured 5/16 inch in 200 feet. This returned to within 1/16 inch of original marks when the load was dumped, which would indicate that the structure is amply stiff.

An inclining experiment conducted on these dredges gave the following results: Empty, *G. M.* = 7.19 as inclined, trans. *B. M.* = 20.4 feet, trans. *C. B.* = 5.53 feet, *C. G.* above base = 18.74; loaded, corrected *C. G.* = 18.14 feet, met. above base = 20.8, *G. M.* = 2.66.

No. 16.—The British International Trophy Race of 1908.

BY W. P. STEPHENS.

ABSTRACT.

The modern speed launch which has of late engaged the attention of sportsmen, yachtsmen and engineers, is essentially a French production, representing a certain stage in the development of the gas-engine intermediate between the automobile and the airship. Its origin is due neither to the naval architect, the marine engineer nor the yachtsman, but to the restless and ambitious builders and owners of automobiles, who, about 1900, transferred the lightest and most powerful of their new engines from their proper place in the car to an improvised setting in some sort of a launch hull. The international auto-boat race of 1908 is noteworthy in that it has established an authentic record of a higher speed than ever before made by vessels of under 40 feet extreme length; but what is of far greater importance is the fact that it has demonstrated beyond question the position of the naval architect as the one controlling power in the production of a perfect vessel, even the marine engineer, essential as he is, being subordinate to him.

The race was held in Huntington Bay, Long Island, Aug. 3, over a triangular course 10 nautical miles long, three circuits of the course being made. The entries included two English boats, the *Wolseley-Siddeley* and *Daimler II.*, and three American boats, the *Dixie II.*, the *U. S. A.* and the *Den II.* The *Dixie II.* was first over the line, followed by the *Den II.*, the *Daimler II.*, the *Wolseley-Siddeley* and the *U. S. A.*; the last 41 seconds after the signal. Shortly after the start the *Daimler II.* speeded up and opened several lengths between her stern and the *Wolseley-Siddeley*, passing the first mark only about 20 seconds after the *Dixie II.*; a little later, however, one of her wing engines gave out and she withdrew. The *Dixie II.* covered the first round of 10 nautical miles in 21:35, leading the *Wolseley-Siddeley* by 47 seconds. She covered the second round in 22:16, with the *Wolseley-Siddeley* but 16 seconds astern. On the second leg of the third round she speeded up, finishing the race with a lead of 33 seconds.

On the following day the *Dixie II.* was run over the measured mile course in Hempstead Bay, really 1.1 nautical miles, four runs being made. The mean of the four runs showed a speed of 31.05 knots, or 35.75 statute miles. On Aug. 20, 21 and 22 the *Dixie II.* raced as one of the challengers for the A. P. B. A. Gold Cup over the Chippewa Bay course on the St. Lawrence River, two rounds making 30.10 statute miles. She won the three races, all run in rough water; the highest speed made being an average of 31 knots in the second race.

While the success of the *Dixie II.* was due to the harmonious efforts of the designer, builder, captain and engineer, the fact that concerns us most to-day is that the master mind was that of the naval architect (Mr. Clinton H. Crane); the others, each doing his own work in his own place, being subordinate to him. It cannot be too strongly emphasized that what has brought success in this instance is not the mere refinement of lines through tank experiments nor the discovery of any new principle that will make possible a higher speed; it is only that, for once at least, the naval architect has been free to exercise the proper functions of his office; to determine from his knowledge of the rules and conditions of this special case the type and general dimensions best suited to them; to select freely, according to his knowledge of the laws of naval architecture and his personal experience with vessels in service, the vital elements of the design, the displacement, freeboard, location of the centers and distribution of the weights; to draw upon his general knowledge of marine engineering for a horsepower fitted to his hull, and then to utilize his special knowledge in the drafting of the lines.

No. 17.—Transportation of Submarines.

BY NAVAL CONSTRUCTOR W. J. BAXTER, U. S. N.

ABSTRACT.

It having become necessary to transport two submarines, length over all 64 feet 9 inches; diameter, outside of hull, inside of bilge keels, 11 feet 10 inches, it was decided to send from the Navy Yard, New York, to a distant port on a collier, length over all 322 feet, breadth extreme 43 feet, depth molded 23 feet 1¾ inches, normal load draft 19 feet 7½ inches, and launch them from the collier's deck upon arrival at the port of destination, after which the collier was to be ready to immediately assume her ordinary duty.

The necessary stability calculations, to determine: first, the stability conditions when the collier was properly loaded with a large cargo, and with the submarines on board; the various changes in water ballast which should be made during the voyage to compensate for the changes in trim and draft which would follow the consumption of coal en route to ensure that, under all possible conditions of weather, the collier would be kept in proper trim, with ample stability; and the provision of suitable additional local strength to prevent any possibility of racking or straining the hull, so as to cause leakage through seams, butts and rivets, or of straining the submarines. This having been satisfactorily determined, the actual working plans were next prepared; but before they were finally completed, a model of a submarine to three-quarter inch scale was made and given a metacentric height corresponding to that of the actual submarine at the time of launching; various model cradles were also tried, and the model was launched into a wooden tank of suitable size to prevent reflex wave action, under various conditions of trim, and under various conditions of inclination of the collier, and, finally, in imitation of the gradual change in the slope of the ways which would occur owing to the shifting of the weights during the launching period. These experiments having fully confirmed the calculations, working plans were prepared with a view of placing these two submarines on the starboard and port sides, respectively, of the collier's after-well deck, their midship sections being approximately abreast the mainmast,

and their axes being placed directly over the outboard coamings of the coaling hatches, and parallel to the bottom of the keel.

The launching ways and the tripping devices were of practically the same type as those usually used in launching sideways, with this difference: that four launching ways and cradles were used, two at each end, each pair being so tied together that they acted as sleds, and, between them, three supporting chocks were located over bulkheads and beams beneath. The entire weight of each submarine was therefore carried on four heavy cradles and three solid chocks, the cradles being so arranged that they would carry their proportion of the weight. The three supporting chocks were so arranged that, at the time of launching, they could be wedged to take all the weight, and permit the removal of the cradle sleds for the purpose of lubricating the ways; they were afterwards to be replaced, and the chocks then knocked down, leaving the entire weight of the submarine on its two cradle sleds, ready to let go, by cutting one rope, at the proper time.

The submarines themselves were prepared for shipment by removing all small portable fittings, the storage battery, and the rudders; the engines were suitably shored, to prevent any racking while in transit; the propellers were not removed.

LARGE GAS ENGINES FOR SHIPS.

BY E. N. PERCY.

Gas producers for use on shipboard have been pretty thoroughly discussed, also the limits of sizes for gas engines, but all of these arguments have presupposed the present type of slow speed, high initial pressure, explosive engine. While the gas producer is getting to be fairly well understood, let us see if some more suitable type of engine could not be considered. We will assume a two-cycle, three-cylinder, fuel-injection engine as the coming type, for reasons detailed below.

Being two-cycle and three-cylinder, it can run at a high speed and be very light, for its power; especially as the fuel-injection type has no explosion, but a constant burning. This type starts easily with compressed air and reverses like a steam engine; in fact, handling just as easily. It would be a very simple engine, without gearing or valves, excepting a very small Stevenson link motion to control the fuel injection and air for starting.

The most successful engines of this type now manufactured exhaust at the end of stroke through a port in the cylinder at one side, receiving the new charge through a port in the opposite side. The new charge is air only, and thoroughly scavenges the cylinder, some of it even escaping to make a clean scavenge; but this represents no loss of fuel, and a loss of power so small that it could not be measured. After compression, on the beginning of the return stroke, fuel (gas or liquid) is injected and burns, not explodes, until cut off by the valve gear. The fuel is ignited in any one of several ways; i. e., high compression, continuous electric spark, red-hot metal or torch. The certainty and the control of these features will be apparent to any one, and, moreover, they are borne out in practice. In addition to this, we may borrow the idea from a well-known make and use the latent heat of the jacket water by letting it boil in the jacket, and after a pressure has accumulated, inject the steam into the cylinder with the fuel, where it acts in such a manner (as we know from fact, not theory) as to increase the economy and saves most of the jacket heat. The engines now on the market, made along the above lines, hold the records for economy, without exception, and in two instances have exceeded the ordinary engine by half; i. e., they take half the fuel. It might be well to notice here that the fuel-injection type of gas engine can be governed as accurately as a steam engine; in fact, could be governed by a steam-engine governor, fly ball or shaft.

Assuming this to be one of three engines, driving a large triple-screw ocean steamer, let us consider the auxiliaries. First, there should be an independent gas engine, driving a small air compressor for starting purposes. There should be a large reservoir, holding considerable air, sufficient for several manoeuvers, and a smaller reservoir, in case it has to be filled quickly for use. There will be an independent electric-light plant, run by small gas engines, to furnish power and light, and possibly ignition for the main engines. As it is very desirable to have the bearings and pins of a large marine engine perfectly accessible at all times, we could not expect to compress the charge of air in the crank chamber, since it is to be all open. Instead of adding a huge low-pressure blowing engine, we will merely have a small high-pressure electric blower, as two or three pounds is sufficient pressure. The fuel pump or small compressor, if gas is used, will be coupled direct to the main shaft, as the fuel is never needed until after the engine starts, but if the engine is very large, a small drum to store the gas under pressure may be used in conjunction with a separate auxiliary compressor, insuring always a supply of fuel to start on.

All the auxiliaries should be electrically driven, not only to do away with piping, but so that they can then be run from the dock by merely connecting with the dock power house, leaving the ship clear for repairs. All pumps will be electric centrifugal, as this type has been successfully used for feed pumps, and indeed for hydraulic pressures as high as 700 pounds per square inch.

With a ship of this description, very few firemen would be needed. There would be no more "cleaning fires," no more boiler repairs and no more stuffing boxes. The ship could be started at a minute's notice, without waiting to "steam up," or even "warm up." Instead of compressed air, it might be found advantageous to have a large dynamo on the forward end of each main-engine shaft for developing power for the auxiliaries, the dynamos being used as motors, instead of compressed air for getting under way.

Steam from the jackets, or even generated in a small boiler, could be used for heating the ship.

The writer believes that it will be very hard indeed to increase the power of marine gas engines of the explosive type, but the fuel-injection type can be designed very much as steam engines, because their initial pressures are not high (excepting the compression-ignition type) and there are no explosions, hence no necessity for the enormously massive construction of the aforesaid type.

Furthermore, gas-engine designers do not seem to appreciate the advantages of multi-cylinder types, combined with high speed and several units; and naturally, when they try to put large power, and explosive power at that, in one large cylinder, that cylinder has to be very massive and very well supported, and experience shows the repairs to be more on this type than on a multi-cylinder engine of the same power and moderately fast speed, particularly if the latter engine is a single-action two-cycle engine, because the pressures are all in one direction, and there is no knocking of parts, no matter how loose they are.

The preparations which are being made in the Belfast yard of Harland & Wolff for the building of the new White Star liners *Olympic* and *Titanic* of 60,000 tons displacement, include the construction of two new 1,000-foot building slips, over which an immense double gantry is being erected, the erection of a large floating crane and the installation of a modern electrically-driven hydraulic and pneumatic plant, at a total cost of £400,000 ($1,946,600). This work will be completed in January.

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Discussion of the Papers Read before the Society of Naval Architects.

As this meeting was not held until after our date for going to press, we are unable to publish a resumé of the discussion which took place at the meeting, as has been our custom in the past. Several important points were brought out, however, which are worthy of comment.

Three of the papers dealt with the important subject of trials of vessels, one being of a general nature, setting forth the practical difficulties of obtaining reliable trial data from merchant ships; and the other two presenting data taken from tests of turbine-propelled ships. Of the latter, one contained the results of a six-hour test under service conditions on a merchant ship, and the other, data obtained on the regular preliminary and final official acceptance trials of a scout cruiser; this data, of course, being obtained under the usual conditions which govern trial trips in the navy.

It rarely happens that the results of speed trials of warships are open to question. The trials are always carefully made under the best of conditions by a corps of trained observers, and include, of course, progressive trials over the measured mile, a four-hour run at maximum speed, and one or more twenty-four-hour endurance runs at lower speeds. On the other hand, it is seldom that speed trials of merchant vessels are as carefully made. Although there are four Government trial courses on the Atlantic coast and two on the Pacific coast available for shipbuilders and owners, frequently the only tests attempted on merchant ships are those made under service conditions, where little opportunity is given the observers to make their observations in a satisfactory way, and, therefore, the refinements and completeness which can be obtained in the case of naval vessels cannot be expected.

This point is well illustrated in the paper already mentioned, which described a six-hour service test on a merchant ship. This test was made on one of the regular runs of the vessel, and was necessarily of short duration. Due to the deplorable reluctance of the builders of the vessel to make public the details of the hull and machinery and the impossibility under the exigencies of a service test of employing the most accurate and refined methods of measuring the coal and water consumption, the results, even though worked out under the personal supervision of a well-known authority on marine engineering, were subject to considerable criticism and doubt.

The method of obtaining the amount of coal burned by counting the number of buckets is, of course, open to question, but, in this particular case, a valuable check on the results was available in the records of the steamship company for the average coal burned per trip on the steamer for the entire season. From May 11 to November 8 the ship made 178 trips between Boston and New York, partly over a course 298 nautical miles long, and partly over a course with a length of 326 nautical miles. The average coal burned per trip was 106.5 tons, 12.5 tons being estimated as the consumption while in port. This figures out as 7.1 tons per hour, which checks exactly with the estimate obtained in the test.

The equivalent evaporation of the boilers from and at 212 degrees, figured out at 11.2 pounds, which, while not impossible is exceptionally high, and as many instances are on record where water meters have been known to give erroneous results, sometimes with a percentage of error as high as 17, the question was at once raised not only as to the accuracy of the determination of the coal consumption, but also as to the reliability of the water meter. In the present case, the meter was especially designed to handle water at the temperature of the feed, and careful calibration under the exact conditions existing during the test showed a difference of only one-half of one percent before and after the test, so that the figures for the water consumption cannot very well be doubted.

The value obtained for the shaft horsepower of the

turbines was criticised mainly because the value for the modulus of elasticity of the shafts was assumed instead of being determined exactly. This would not be admissible if there were any reason to believe that the modulus of elasticity would vary within wide limits. But it is well known that for different pieces of steel of the same properties this value varies very little. The value 14.76, obtained for the steam consumption of the turbines per shaft horsepower per hour seems, of course, at first sight to be somewhat larger than might be expected. It should be noted, however, that in this value no account has been taken of the four percent priming in the steam. Taking account of this priming and reducing the steam consumption per shaft horsepower to steam consumption per indicated horsepower on the assumption that the shaft horsepower is between 85 and 90 percent of the indicated horsepower, we find that the steam consumption per indicated horsepower per hour would be about 12.5 pounds, which is by no means a bad showing for either a reciprocating or turbine engine.

The value of model tank experiments in settling doubtful points relating to the resistance of ships was again demonstrated this year by the results of experiments performed by Naval Constructor Taylor at Washington and Professor Sadler at the University of Michigan. The influence of a change in the shape of the midship section of a vessel upon its resistance has hitherto aroused much speculation. As a result of Mr. Taylor's experiments, designers can now be reasonably sure that, for a midship section of definite area, the shape can be varied without affecting to a great extent the resistance of the vessel. The few experiments which had hitherto been made on models with a fine midship section and a full midship section seemed to indicate that the full model would drive easier. This has now been shown to be true, and a section with a flat bottom and deep bilge can be expected to give better results than one in which the draft is increased and a comparatively fine section used. Of course, it is not always expedient to use a broad section, for in sea-going ships increased draft is frequently necessary to insure proper sea-going qualities. In connection with a broad, full midship section hollow bow lines seem to be an advantage.

Hollow bow lines do not necessarily cause excessive pitching, because that is largely due to synchronism. In one particular case where two models of English warships were tested, one with hollow bow lines and the other with straight lines, it was found that an advantage of one knot in speed could be gained by the use of the hollow lines.

Attention was called by one paper in particular to a type of ship which is distinctly American. This paper described in detail the latest addition to the fleet of shallow-draft paddle steamers operating on inland waters. This boat, which is only 455 feet 2 inches long over all and 55 feet wide, with a maximum displacement of 6,410 gross tons, at which the draft is 15 feet, has accommodations for as many passengers and as much freight as the giant Cunard steamship *Lusitania,* which has a gross tonnage of 30,822, a maximum draft of 37½ feet, and is 785 feet long over all, with 88 feet beam.

The problem of building a boat which is in reality a six-story floating hotel for such a capacity and upon such small draft is certainly noteworthy. Nothing is sacrificed on boats of this type which will add to the comfort and safety of the traveling public, so that, in completeness of equipment and luxurious appointments, these ships compare favorably with the finest ocean liners. The problem of propulsion in ships of this kind is of the utmost importance, and, for reasons of both economy and convenience, the paddle steamer has never been superseded by propeller-driven ships, even though the introduction of steam turbines has tended to reduce the diameter of propellers and decrease the vibration.

Fuel for Internal Combustion Engines.

Many tests of internal combustion engines have been made, but a large majority of them have been made by private concerns for a specific purpose, and the results are not generally available. Furthermore, as is generally recognized by those familiar with gas engines, especially those operating with gasoline (petrol) as a fuel, the conditions influencing engine performance are so numerous, and vary to such an extent, as to make the value of off-hand comparison very limited, and, oftentimes, misleading. Exact comparisons are only possible under identical conditions, or when the actual differences in all conditions that influence the results are exactly known. With a view to obtaining comparative results, which would be of value concerning the operation and design of gasoline and alcohol engines, the Technologic Branch of the United States Geological Survey has recently made an exhaustive investigation, the results of which are published elsewhere in this issue.

The work was taken up to investigate the characteristic action of fuels used in internal combustion engines, with a detailed study of the action of each fuel (gasoline and alcohol) as governed by the many variable conditions of engine manipulation, design and equipment. These variables, so far as possible, were isolated; their separate and combined effects were determined and worked out under practical operating conditions, leading up to the conditions required for minimum fuel consumption. The results show the saving that can be obtained for conditions over maximum consumption and also establish a definite basis of comparison under conditions most favorable to each fuel. This latter is a point of much commercial interest, and, undoubtedly, this study of the comparative action of gasoline and alcohol will be of great service in solving some of the general internal combustion engine problems where other than liquid fuels are used.

Progress of Naval Vessels.

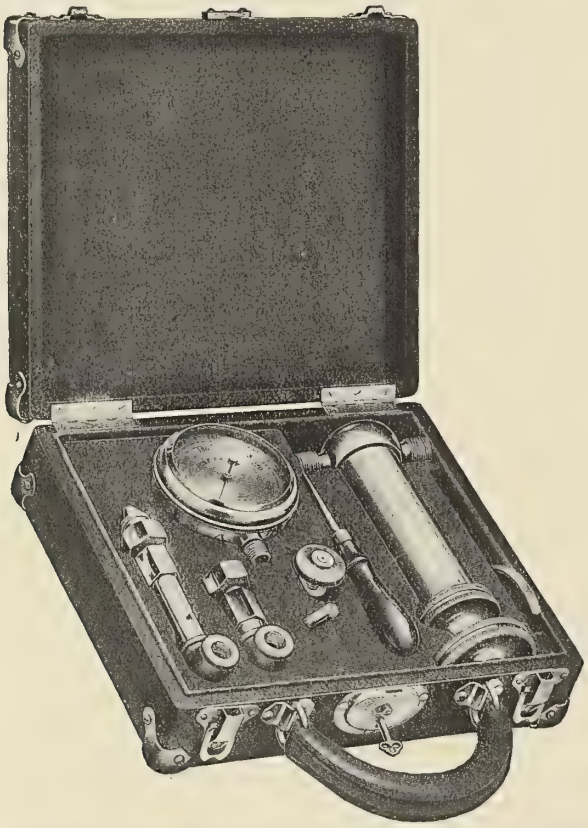
The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Tons.	Knots.		Oct. 1.	Nov. 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons.....	63.2	65.9
Michigan ...	16,000	18½	New York Shipbuilding Co..	71.4	74.9
Delaware ...	20,000	21	Newp't News S.B. & D.D. Co	44.2	50.3
North Dakota	20,000	21	Fore River Shipbuilding Co.	54.2	58.8
TORPEDO BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons.....	53.8	57.5
Lamson	700	28	Wm. Cramp & Sons.....	52.1	56.2
Preston	700	28	New York Shipbuilding Co..	49.1	52.0
Flusser	700	28	Bath Iron Works.....	26.1	33.0
Reid	700	28	Bath Iron Works.....	26.0	31.6
SUBMARINE TORPEDO BOATS.					
Stingray	—	—	Fore River Shipbuilding Co..	58.7	62.3
Tarpon	—	—	Fore River Shipbuilding Co..	57.3	60.3
Bonita	—	—	Fore River Shipbuilding Co..	55.5	57.8
Snapper	—	—	Fore River Shipbuilding Co..	55.7	56.5
Norwhal	—	—	Fore River Shipbuilding Co..	50.9	52.3
Grayling	—	—	Fore River Shipbuilding Co..	50.7	52.0
Salmon	—	—	Fore River Shipbuilding Co..	50.3	51.3

ENGINEERING SPECIALTIES.

The American Inspector's Outfit.

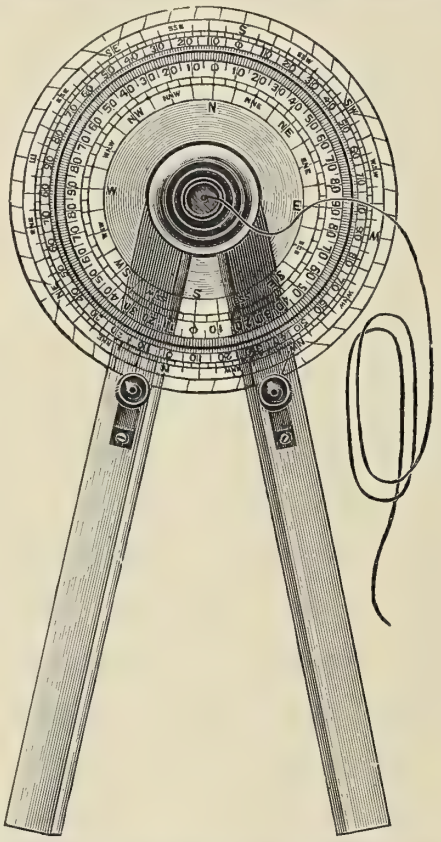
This outfit, which is manufactured by the American Steam Gauge & Valve Manufacturing Company, Boston, Mass., consists of a 3-inch test gage of 300 pounds' capacity, a screw test pump, a hand puller, a hand set, pliers and screwdriver. All



of the instruments are nickel-plated and packed in a morocco velvet-lined case, the total weight of the outfit being about 6 pounds. Thus it can be easily carried about and is always ready for use. The outfit is especially designed for a portable testing apparatus for testing locomotive and other boiler gages, and meets the requirements of government, power plant and boiler insurance inspectors.

Captain Ashe's Patent Course and Position Finder.

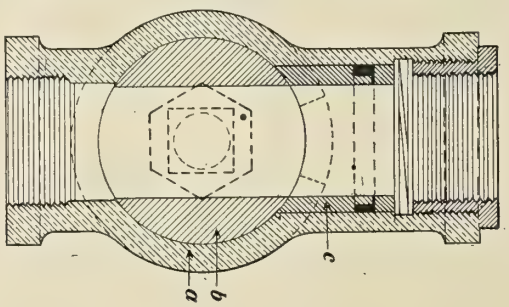
Accuracy, simplicity and strength are fundamental requirements for navigators' instruments. The fulfillment of these requirements has been sought in Capt. Ashe's patent course and position finder, manufactured by Heath & Company, Ltd., Crayford, London. This is a new chart instrument, by means of which many arithmetical calculations hitherto inseparable from chart work are entirely obviated, while in addition it is useful for the following purposes: Finding the ship's position from course bearings of two or three objects by simple



compass observation; finding the true and magnetic courses simultaneously; finding the course by observation when the deviation is known, and laying off the course without reference to the compass-croses on the chart. It is claimed that the simplicity of the instrument and the ease with which it can be manipulated render errors in courses or bearings impossible, and also permit a frequency of use which would not be justified with more elaborate and more expensive instruments. The instrument is transparent throughout, so that all details of the chart can be observed. It has been invented by a practical mariner and is intended for the use of practical mariners.

The Caskey Valve.

A new and unique valve, which can be used for several purposes, has recently been placed on the market by the Caskey Valve Company, Philadelphia, Pa. The valve may be used as a boiler blow-off valve, a hydraulic-operating valve, or as an air and relief valve. It is designed to maintain a pressure up,



to 10,000 pounds per square inch, and becomes more effective as the pressure is increased. Of course, the valve is made in different weights and in slightly different form for the various purposes for which it may be used; but in all cases the valve is a straight-through valve, with no pockets or recesses. Furthermore, there are no stuffing-boxes, valve seats and, consequently, no fluttering action. The illustration shows a sectional view of the boiler blow-off valve. It consists of the body, *A;* straight plug, *B;* the bushing, *C*, ground to fit the plug, *B;* while to assure contact between *B* and *C* a spring is provided. The valves are made in sizes ranging up to 6 inches.

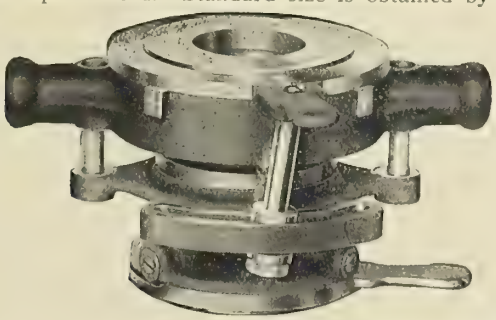
The B. & S. 15-Degree Angle Motor Boat Wrenches.

The Billings & Spencer Company, of Hartford, Conn., are putting out a new set of 15-degree angle double-end wrenches that are adapted to use on motor boats, etc. These wrenches,

seven in number, ranging from 7/16-inch milled opening to 1¼ inch, are put up in serviceable waterproof kit bags of duck or drill, and are furnished either semi-finished or with a full case-hardened finish. This should prove a very convenient set of tools for general use aboard a motor boat.

Oster Matchless Die Stock.

The Oster Manufacturing Company, of Cleveland, Ohio, has recently brought out a new tool for threading pipe. This is a die stock for pipe ranging in size from 1 to 2 inches in diameter. The dies are controlled by a cam, which, following the lead screw, is driven by a guide post at the exact taper for pipe thread. The dies automatically recede to make a standard taper thread. Standard size is obtained by setting



the guide post in position to indicate the size as graduated on the face plate, and locking the die solid by the nut at the bottom of the post. This setting remains unaltered as long as duplicate threads are desired. Over or under-size threads, or crooked threads, can be obtained if desired, and one set of dies may be used for all sizes or a set for each size, as the operator prefers. An important feature is the universal gripping chuck, which adjusts to all sizes by revolving a handle; this takes the place of all bushings and makes the stock entirely complete within itself. There are no loose parts, and as the gripping jaws are of tool steel, hardened, it is claimed that they will effectually withstand the strain and wear.

TECHNICAL PUBLICATIONS.

The Mechanical Engineering of Power Plants. By Frederic Remsen Hutton, E. M., Sc. D. Size, 5¾ by 9 inches. Pages, 825. Figures, 697. New York, 1908: John Wiley & Sons. Price, $5.

A former edition of this book, issued in 1897, embodied the study and experience of the author, gathered during the previous twenty years, and brought together for teaching purposes. Due to the fact that the years since then have been a period of great and rapid progress in the development of the power plant, and of engineering departments pertaining to it, it has been found necessary to rewrite the entire book. The present edition is the result of this work.

The new features which are specially noteworthy are the analysis of the power plant and its diagram; the separation of the simple and complex phases of this problem; the treatment of the steam pipe as an element of co-ordinate importance in the plant with the boiler and engine; the chapters on the auxiliaries as distinguished from the essentials; a discussion of the steam turbine; the establishment of the philosophy of the expansion of the elastic medium as the basis for the valve gear, the governor, the condenser and the compound engine. Statistics and tables have been very largely excluded, it being the intention of the author that engineers' pocketbooks should be consulted for this information.

The main headings under which the subject is treated are the quantitative basis of the steam-power plant, leading up to the cost of a horsepower; a comprehensive treatment of the steam boiler, including all its accessory apparatus, care and management, piping, etc.; an equally complete treatment of the engine, including a description of the design of the ordinary reciprocating engine, the rotary engine, steam turbine, valve motions, governors, condensers and auxiliary apparatus. Considerable space is given to the care, management and testing of boilers, engines, etc.

Taken all in all, this is undoubtedly one of the most complete and valuable works covering the steam-power plant which is available for engineers to-day.

The Temperature-Entropy Diagram. By Charles W. Berry. Size, 4¾ by 7¼ inches. Pages, 299. Figures, 109. New York, 1908: John Wiley & Sons. Price, $2.

Students of thermodynamics are usually puzzled about the true significance of entropy, and frequently fail to realize the usefulness of the temperature-entropy diagram. This is the only book, so far as we know, which is devoted exclusively to the subject, presenting it in a clear and available manner. The subject matter has been gathered in an effort to bring together in logical order certain information concerning the construction, interpretation and application to engineering problems of the temperature-entropy diagram which is not readily accessible. An exhaustive treatment of the subject has not been attempted, but the graphical presentation of the subject given is such as to make clear the fundamental principles of thermodynamics.

The present volume is a revised edition, the first volume having appeared in 1905. A graphical method of projecting from the pressure volume into the temperature-entropy plane has been elaborated for perfect gases and its application to hot-air engines and gas engines given. The various factors

affecting the cylinder efficiency of both gas and steam engines are thoroughly discussed. One chapter is devoted to the thermodynamics of mixtures of gases and vapors, and another to the description and use of Mollier's "Total Energy-Entropy Diagram."

Fighting Ships, 1908. Edited by Fred T. Jane, Size, 12 by 7½ inches; numerous illustrations. London, E. C., 1908: Sampson, Low, Marston & Company, Ltd. Price, 21s.

Since this is a book which has appeared annually for the past eleven years, its general scope and arrangement are well known to naval men in nearly every country. Aside from the changes which occur in the subject matter, as the result of changes in existing warships and the addition of new ships to the various navies, several important changes have been made in the manner of compilation and methods of presenting data. In the previous volumes, while official classifications have been generally adhered to, the ships in any particular class have been roughly arranged according to fighting value. In the present volume this arrangement has been discarded in favor of a strictly chronological one. That this is a fair classification is evident when it is considered that recent ships, even if of inferior displacement and gun power, are of much greater value, due to the modern fittings, gun mountings, electrical gear, etc., which are installed. Furthermore, defects which exist in older ships have been remedied in the newer types. Official classifications have been strictly adhered to, for the reason that while some ships, such as the British *Invincible* class, are more powerful than most battleships, yet they have not been classed as such any more than the *Dreadnought* has been classed as a cruiser, as she might well be, because she has a higher speed than many cruisers.

Part I., which comprises all important data regarding every warship in the world, together with illustrations and diagrams, is arranged by navies in the order of their importance. This is the customary arrangement of the book, but this year the order has changed, so that while Great Britain and the United States are first and second, respectively, France has dropped to fifth place, with Germany and Japan third and fourth, respectively. Italy is placed sixth and Russia eighth.

Among the articles in Part II., dealing with various phases of warship construction, is an article describing a new type of conning tower for large battleships by Commander Hovgaard, of the Royal Danish navy, now professor of warship design at the Massachusetts Institute of Technology; an article on the future 20,000-ton warship, by Col. Cuniberti, of the Royal Italian Corps of Naval Constructors, who may be called the father of the *Dreadnought* type of battleship. Marine turbines are thoroughly discussed, drawings being shown of both the Parsons and Curtis types. Various types of watertube boilers used on warships are also illustrated and described, and in an article on the "Progress of Warship Engineering," such subjects as turbine performances, torsion meters, steam generators, superheating, liquid fuel, internal-combustion engines, ventilation, valves, torpedo-propelling engines and wireless telephony are thoroughly considered.

Silhouettes of merchant ships over 5,000 tons gross and 18 knots speed are given, as well as a merchant index, in which is listed every merchant ship of any importance which might be met upon the high seas.

Valve Setting. By Hubert E. Collins. Size, 6 by 9 inches. Pages, 209. Figures, 200. New York, 1908: Hill Publishing Company. Price, $2.

Practical instructions in the setting of valves for all kinds of engines are a necessity for supervising, operating and erecting engineers. Such instruction, secured principally from the builders or erecting men, who are familiar with the practical work involved, has been brought together in this book, the subject matter of which has been published in various issues of *Power*. Careful discussion of the slide valve is given in the first few chapters, as the principles involved in setting slide valves are fundamental. This is followed by a consideration of the Meyer cut-off valve and various types of Corliss valve gear. General rules are given for finding crank and eccentric centers which can be applied to any make of reciprocating engine. It is intended that after the reader has mastered the details in the first few chapters of the book, he will be able to read understandingly the remaining portions, which are devoted to a description of the valve gears of the most important commercial types of steam engines now on the market. In the last two chapters valve setting for duplex pumps and air compressors is described.

Patents as a Factor in Manufacturing. By Edwin J. Prindle. Size, 5 by 7½ inches. Pages, 134. New York, 1908: *The Engineering Magazine*. Price, $2.

This volume is not intended to give the inventor or the manufacturer sufficient information so that he may act as his own patent lawyer, but it is intended rather to convey an idea of the nature of a patent, the protection it may afford, the advantages it may possess for meeting certain commercial conditions, the safety which may be secured in relations between employers and employees, and the general rules by which the courts will proceed in upholding a patent and in thwarting attempts of infringement. It is intended to give the inventor or manufacturer a grasp of the fundamental principles, so that he may proceed rightly in the early steps which are usually taken before the advice of counsel is secured, after which it is pointed out when and where it is necessary to call in expert legal advice. The author of the book, as the result of wide practice, both in mechanical engineering and in patent law, is in a position to appreciate the important points of this subject, and place them before the reader in such a way that the precautions which should be taken in the preliminary steps, and the rules and principles which should be followed, are made clear. The book includes the following chapters: Influence of patents in controlling the market; subject, nature and claim of a patent; what protection a patent affords; infringements; patenting new products; patent relations of employer and employee, and contests between rival claims to an invention.

QUERIES AND ANSWERS.

Questions concerning marine engineering will be answered by the Editor in this column. Each communication must bear the name and address of the writer.

Q. 419.—The figures which you published in your January, 1908, issue in connection with a test on the steamship *Governor Cobb* have been questioned at a recent meeting of the Marine Institute. Are these figures thoroughly reliable? B.

A.—The figures for both the steam consumption of the turbines and the water evaporation of the boilers were severely attacked at the meeting of the Society of Naval Architects, at which the paper was presented. The builder of the boilers and vessel questioned the steam consumption of the turbines, as he claimed the evaporation of the boilers was too high. The evaporation figures out, allowing for the proper percent of moisture, 10.85 pounds of water per pound of coal from and at 212 degrees. The boilers of the *Lusitania* show an evaporation of 11.1 and 10.9 pounds, and in a recent test made on the steamship *Harvard,* where every possible precaution was taken, a slightly higher figure was obtained. Considering that the *Cobb* used cold air for forced draft, the 10.85 seems to be too high a figure. The method of determining coal on that test, by counting buckets, is, of course, always open to question, and is undoubtedly subject to an error.

Just what this error may be you can judge for yourself, but there is every reason to believe that determining coal in this manner will give too small a figure. There is every reason to believe that the water meter used in determining the feed-water gave very accurate results. The meter was calibrated by the manufacturers before leaving their works, and was calibrated at the Institute of Technology after the test was made. The fact that a curve of meter readings, plotted at ten-minute intervals, follows exactly the fluctuations of power, is a pretty strong indication that the readings of the meter were not sensibly in error. The only other thing that could affect the water rate would possibly be a leak in the feed-water heater, but when it is remembered that this test was run on the second round trip, after the boat had been thoroughly overhauled for its season's work, and the feed-water heater pronounced in perfect condition, the chances that it sprung a leak in so short a time are rather remote. If the determination of coal on the *Governor Cobb* were, let us say, 4 percent too low, the equivalent evaporation would then have been about 10.4 pounds, a good but by no means impossible figure, the ratio of heating to grate surface being over 37 to 1.

The figures for steam consumption do not appear to be at all favorable from an economical point of view, but it is a well-known fact that the *Cobb* in actual service uses vastly more coal than was expected. L.

Q. 420.—Are licensed officers required on pleasure motor boats? What equipment is required on such boats? NEW YORK.

A.—No licensed officers are required on any pleasure motor vessel, no matter how large she may be, and if the boat is under 16 tons it does not have to be documented at the custom house. Licenses are held by the operators of a good many pleasure motor boats, who wish occasionally to earn a few dollars by taking passengers for hire, taking parties to the Fishing Banks, or leasing their boats for a day or two. Motor-driven fishermen, oystermen, etc., are also exempt from inspection, and do not require licensed officers. This creates a great danger in passenger business, as some boats are registered as oystermen, but never do anything but passenger business, and probably a large majority of the boats which ordinarily are oystermen take out fishing parties for hire on Sundays and holidays. In the Second District upwards of 3,000 licenses have been issued to operate passenger motor boats of 15 tons or less, while there are not in this district 100 motor boats documented as passenger boats, showing that the great majority of these operators call their boats either pleasure, oyster or fishing vessels, although a great many of them are not documented at all.

As to the equipment which is required on these boats, there is no law which in so many words says that everybody aboard such pleasure craft shall be provided with a life preserver, but as this is such a common-sense requirement, and is required on all commercial vessels, and as yachts are made subject to certain portions of Title LII. of the Revised Statutes, and as they are given certain privileges by those statutes, it has been held by many people that the inference is plain, that on pleasure craft, as well as commercial craft, there should be a life preserver for each person on board. Where such craft have air cushions, or other devices equivalent to life preservers, these might be considered ample protection.

The lights, fog-horn, bell and whistle are all required by act of Congress. In very small motor boats it is impracticable to carry separate side lights, and the supervising inspectors have held that the red, white and green combination lantern, secured on the bow to the jackstaff, or about that position, answered all legal requirements. Of course, separate side lights are better where they can be carried. Every such boat should carry a white lantern as an anchor light. This is a very essential article of equipment, and fully three-quarters of the small yachts fail to comply with this law. It is, of course, not essential that when yachts are all moored together in their regular harbor they each have up these lights, but the law does not provide for exceptions.

As regards a power fog-horn, on all such craft a 25-cent tin horn is all that the law requires. A bell of some sort that can be heard in a fog for a distance sufficient to enable a steamer to clear the anchored craft is required by act of Congress. The practice in regard to the whistle is not uniform. Passenger motor vessels require a mechanical whistle, and on such small craft the small tank where the air is compressed by the motor and released by a lever is advocated. More complicated machines have been fitted up for yachts, but they have usually not proved reliable. On small pleasure motor craft the automobile whistle, or "honk," where the blast is given by squeezing a rubber bulb, would probably comply with the law, while on larger vessels the compressed air tank and lever should be supplied. H.

Notice.

In order that none of our readers may be misled, we wish to call attention to the communication, "Steaming Radius of Scout Cruisers," published on page 454 of our October, 1908, issue, signed "Parsons." This communication was not written by the Hon. C. A. Parsons, or anyone connected with the Parsons Marine Steam Turbine Company, Ltd., the signature being simply a nom-de-plume.

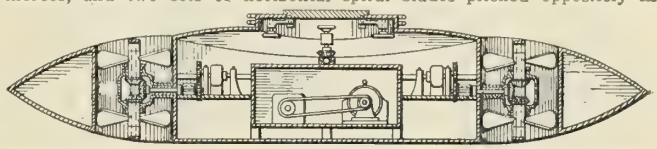
SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

896,361. SUBMARINE BOAT. JOHN M. CAGE, DENVER, COL., ASSIGNOR, BY MESNE ASSIGNMENTS, TO THE SUBMARINE NAVIGATION AND MANUFACTURING CO., DENVER.

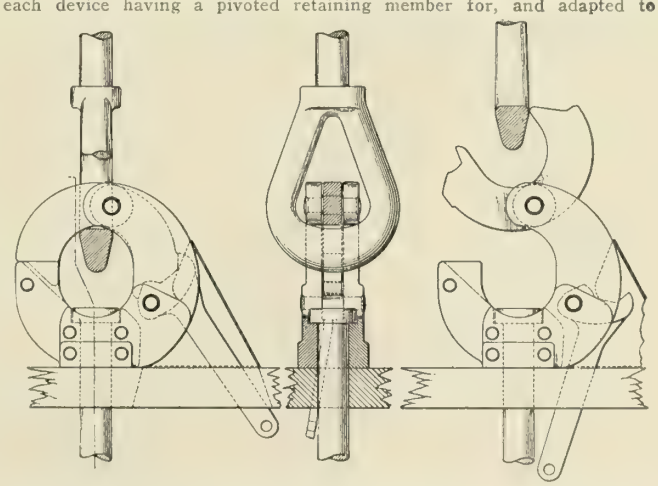
Claim 2.—A submarine boat having a vertical tube adjacent each end thereof, and two sets of horizontal spiral blades pitched oppositely in



respect to each other and arranged within each tube to move in opposite directions about a common vertical axis, and removable covers for each end of each tube. Two claims.

899,072. RELEASING DEVICE FOR BOATS. WILLIAM G. RANDLE, CHESTER, PA.

Claim 3.—In combination, a boat having seats and provided at each end with a davit connection securing device projecting above said seats, each device having a pivoted retaining member for, and adapted to

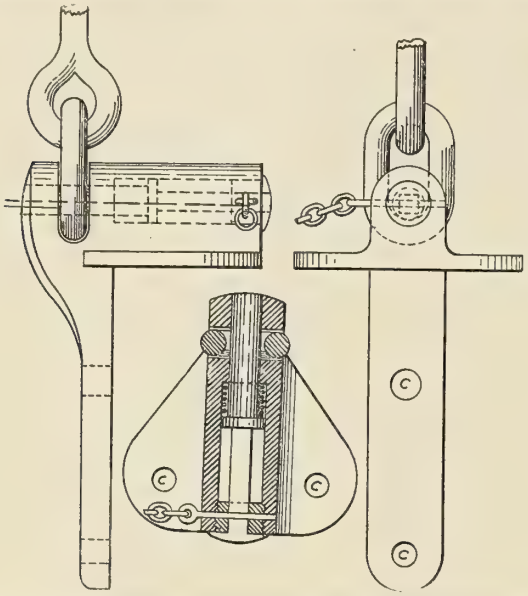


directly engage said connection and a locking dog having a portion projecting below the seats, normally holding said member in the retaining position, but movable to release said member, said member and dog having cam surfaces which directly engage to move the retaining member out of the retaining position, when the dog is moved to release the re-

taining member, and means located beneath the seats and engaging the portions of the locking dogs projecting below the seats for simultaneously moving the locking dogs to release the retaining members controlled by them. Five claims.

898,004. LIFE-BOAT DETACHING DEVICE. JOSEPH PREVOT, GREEN BAY, WIS.

Claim.—A boat-releasing device comprising a metal hanger, bolt plates, a barrel portion, a vertically-disposed slot, a link in said slot, a spring-actuated plunger provided with a cylindrical portion engaging



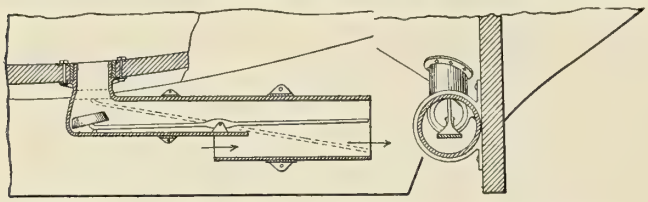
said link, and said plunger also provided with a square portion, a bushing with a square hole threaded into the rear portion of said barrel, and a holding pin for holding the parts in position. One claim.

895,648. ANCHOR OF SUBMARINE BOATS. MAXIME LAUBEUF, PARIS, FRANCE.

Claim.—A submarine or submersible boat having a tube vertically disposed through the hull, having an enlarged lower end for receiving the flukes of an anchor, a chain movable within said tube, a stopper for engaging the chain in the conduit and movable from the interior of the boat, an anchor connected with the chain, and means for preventing the flukes of the anchor touching the hull when the anchor is drawn into the enlarged lower end of the tube. One claim.

899,248. BILGE WATER EJECTOR. LOUIS EBBE, BLAINE, WASH.

Claim 1.—An attachment of the character described, comprising an outlet tube, a tubular body extending beyond one end of said tube and having an outlet at one end and a reduced inlet opening at its other

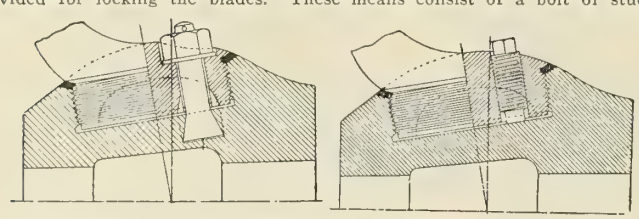


end and below the tube, said tube being disposed to be connected to the bilge of a boat, a valve, and a gravity-operated member for holding the valve normally upon its seat to close the inlet of the tube, said gravity-operated member being disposed within the body, and between the reduced inlet and the outlet thereof. Six claims.

British patents compiled by Edwards & Co., chartered patent agents and engineers, Chancery Lane Station Chambers, London, W. C.

8,880. SCREW PROPELLER. T. EATON, NORWICH, CHESHIRE.

In those propellers, in which the roots of the blades are screwed into the boss and, after adjustment, are fixed in position, means are provided for locking the blades. These means consist of a bolt or stud



passing through the root parallel to its axis, and adapted by its action to press the threads on the root on to the threads in the recess. In one form, the screwed bolt bears on the bottom of the recess; and, in another form, the headed bolt engages in an undercut recess formed in the propeller boss.

8,879. LIFEBOATS. V. COOMBE, BIRKENHEAD.

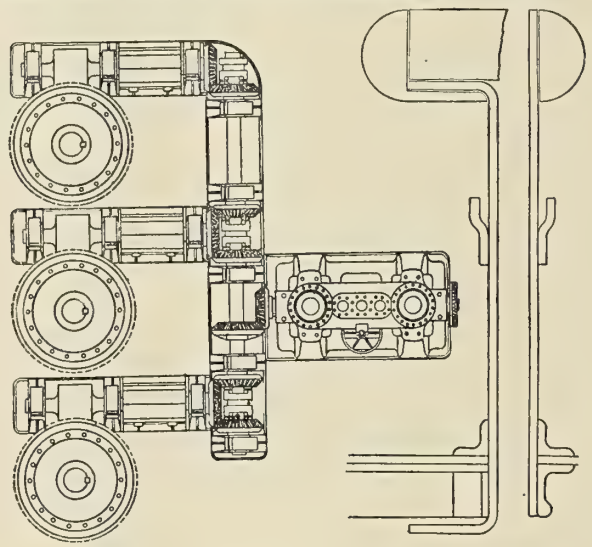
Relates to lifeboats or rafts wherein air chambers or belts of cork are provided on both sides of, and above and below, a platform, which projects beyond the air chambers to form a fender, and consists in providing the air tubes or chambers with battens on both sides to protect the upper and under surfaces. In the reversible lifeboat the platform extends outside the tubular shell, forming a fender. The tubes and platform are fixed together by bolts passing through flanges on the tubes. These tubes are divided into a plurality of compartments by transverse partitions, one or more of these compartments being used for stowing commodities, etc., the others being wholly or partly filled with cork or like material. The boat always floats with the platform above water, and any water that may be shipped discharges through plug holes in the platform.

10,200. MAGNETIC COMPASSES. F. W. CLARK AND KELVIN & JAMES WHITE, GLASGOW.

The journals of the compass bowl and gimbal ring are supported on anti-friction rollers in lubricant filled casings on the gimbal ring and binnacle flange, respectively. Covers keep dust from the casings, which may be supported on springs.

10,295. CAPSTANS. A. KELLY AND C. D. B. HANSEN, GLASGOW.

Three capstans, arranged in a line across the deck of a vessel, are geared to a single winding engine so that they are operated simultaneously; also a fourth hand-operated capstan is positioned so that it can be used to wind any one of the three ropes. Each of the power capstans has a driving connection with a vertical shaft, preferably consisting of a pin disk on the shaft engaging corresponding recesses on the capstan. Each shaft is driven from the engine through a clutch, bevel gearing and worm gearing. Each power capstan is fitted with a suitable friction brake, and one or more of them may be provided with warping drums or whelps for winding steel hawsers.



10,295. 10,801.

10,801. SHIPS' HATCHWAYS. R. I. DUNCAN AND A. ELLIOTT, HARTLEPOOL.

The hatch coaming plates are flanged at the top, and are provided with angle bars, one flange of which forms a molding. The ends of the hatch coamings are not flanged, and the web of the angle bar is cut away when turning the corner, so that the molding flange comes against the coaming.

10,314. SHIPS. C. JEREMIASSEN, PORSGRUND, NORWAY.

Relates to cargo vessels of the type wherein the sides of the hatchways slope from the hatch coamings to the stringer plate, and consists in sloping the ends of the hatchways from the hatch coamings to the deck level. The sloping sides of the hatchways are provided with longitudinal girders, in order to dispense with hold beams and stanchions. Where several hatches are arranged in line the sides are made continuous, and in place of the deck between the hatches a strong transverse connection is fitted. In this construction of vessel the bulwarks may be dispensed with.

10,549. LOADING, ETC. R. BROWN, LIVERPOOL.

A derrick-supported bucket elevator provided with means for breaking and receiving and delivering the material is mounted upon a tender and is employed for loading a vessel from a barge arranged between the tender and the vessel. At the lower end of the elevator frame a shovel or scoop is carried on a crank rotated from the chains. The scoop is provided with a curved guide sliding through a fixed plate, and is so arranged that, as the shovel passes into the material, it assumes a horizontal position and raises a load, which it deposits into the ascending receptacle. The shovel may be flexibly mounted so as to give under a heavy load. In order to give a quicker dipping movement to the shovel, it may be eccentrically mounted on its crank. The scoop carries on its under surface a series of claws designed to bring forward for the next stroke the coal directly behind the scoop.

11,013. STEERING GEAR. A. KELLY AND C. D. B. HANSEN, GLASGOW.

When two steering engines are employed a single controlling gear is adapted to control either engine. The shaft is driven by either or both engines through clutches, and is provided with a worm which engages the worm wheel of the steering shaft. On the main shaft is an additional worm wheel, which engages a worm supported in bearings. A block is rotated by the latter and is moved longitudinally by a hand-operated screwed spindle, and is connected by levers with the shaft which actuates the controlling valves. The spindle is geared to a shaft leading from the steering pedestal, and is provided with a pointer which indicates the position of the steering shaft.